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Design and Optimization of a FSAE Vehicle

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Design and Optimization of a FSAE Vehicle

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MAJOR QUALIFYING PROJECT

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Abstract

The purpose of this MQP was to research, analyze, and supply the frame and various sub-components for the design of the new 2016 FSAE car for competition. The MQP carefully studied the older design and created the new design and major sub-systems while working closely with the future 2015-2016 FSAE MQP team and SAE club members.

A key goal was to improve the performance (handling, acceleration, braking, information gathering, serviceability, and reliability) of the future car. Design and analysis of the sub-system components and the structural frame layout was performed using SolidWorks Simulation. Each sub-system of the car was optimized for integration and performance to ensure that the car will be competitive once built. The car will then be refined, assembled, manufactured and tested for optimization by the 2015-2016 FSAE MQP team.

Executive Summary

Formula SAE (FSAE) is a competition organized by the Society of Automotive Engineers (SAE) for collegiate teams to design, manufacture, and race a prototype Formula style car. Teams operate under the premise that they work for a design firm creating a prototype vehicle for the non-professional, weekend, competition market. To determine the best design, the teams then travel to official competitions where their entries compete in a variety of static and dynamic events.

Our MQP team was tasked to design a car for the FSAE competition in 2016 and manufacture the frame and acquire the engine. To achieve this, the team split into 4 groups focusing on a section of the car: drivetrain, chassis and suspension, brakes and steering, and accessories and electronics.

The drivetrain consists of an engine, transmission, differential, axles, and wheels. A single cylinder engine was selected for its advantages in simplicity and low weight. A 2005 Yamaha YFZ 450 engine was purchased. The engine had a big bore kit installed, improving the displacement by about 10%. To further improve the engine's output, a turbocharger will be installed to boost air intake. Calculations determined that the Honeywell MGT1238 is an excellent option. The engine was mounted onto a test stand to enable testing and work without having the rest of the car available. Many subsystems of the engine, including the oil tank, oil breather box, throttle body, and exhaust were also purchased.

Previous WPI FSAE teams have used a continuously variable transmission (CVT), but selecting a manual transmission broke this trend. While CVTs offer some advantages in performance and efficiency, they also require a substantial amount of tuning and maintenance to obtain the best results. This ran contrary to the project's goals of durability and reliability. In addition to better packaging of subsystems in the frame, selecting a manual transmission allowed many more engines to be chosen as the manual transmissions are dominant amongst the range of engines researched. Although the original engine is carbureted, boosted engines must be electronically fuel injected according to FSAE rules. The engine flywheel was machined to enable position tracking precisely and other components selected, including a fuel pump.

A differential aids turning and handling by allowing the wheels to spin at different speeds. As handling is a significant portion of the dynamic events, it was selected to use a limited-slip differential (LSD) that additionally allows different amounts of torque to be delivered to the wheels. A differential from a Honda TRX300 Fourtrax was acquired for use and a housing designed so it can be adapted to work with a drive sprocket. Axles from a TRX300 ATV were purchased and will be lengthened to fit the drivetrain.

Every year FSAE requires teams to construct a brand new frame. For the new car careful attention was taken to insure the new frame would meet and exceed performance specifications. This included torsional stiffness, weight, and deflection under load. A new 4130 tubular frame was developed with improved driver ergonomics, engine mounting, and control arm placement. The final frame provides increased stiffness over the previous frame while not increasing weight. The frame also provides better rule compliance due to revised roll hoop designs and side impact members.

One of the key components is the suspension. The suspension controls most aspects of tire control. As such careful attention was taken during its development. Several suspension designs were researched and a design matrix was used to make a final selection. The double A-arm SLA suspension provides the needed levels of performance for this application while providing for great adjustability. This design also allowed for different damper actuation methods. The final suspension design provides the required camber, roll center, and damper control. The new pullrod damper system greatly improves the old cars pushrod and direct acting systems. It provides a low center of gravity while providing excellent packaging.

To make sure all developed suspension and frame components would stand up to the rigors of a racing environment a full analysis of the dynamic loads was done. This included 1.5g deceleration, 1.5g lateral acceleration, and a 3.5g bump event. The force was calculated for the wheel loads, suspension loads, and pullrod loads. These values were then used to run SolidWorks FEA simulations to measure the deflection and factor of safety for each component.

The braking and steering systems of the vehicle are critical in determining how the car will perform in competition. Using transfer of energy equations the braking system was simulated and based on the results the components were selected that best matched the car. This ensures that the car will be able to lock all of the wheels with reasonable effort and be able to slow the car in a predictable manner. The steering system of the car includes the uprights and hubs, which are made completely by the team. These parts involve complex design and interact with multiple systems in the car, which is why their accuracy is crucial. The hubs and uprights of the car were tested in FEA to ensure their safety and reliability when they are used on the track. All of the braking and steering components were modeled in a SolidWorks assembly which allows the team to check their function and fitment.

The accessories and electronics of the car were researched in preparation for the following year. They are divided into the following subsystems: cooling system, exhaust, intake, fuel system, and electronics. Various components were chosen and purchased this year while others await decisions that must be made by the following MQP group. In this part of the project, ground work was done to allow for an easier transition in continuation with the rest of the car.

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Overview of Formula SAE Competition

Formula SAE (FSAE) is a competition organized by the Society of Automotive Engineers (SAE) for collegiate teams to design, manufacture, and race a prototype Formula style car. Teams operate under the premise that they work for a design firm creating a prototype vehicle for the non-professional, weekend, competition market. To determine the best design, the teams then travel to official competitions where their entries compete in a variety of static and dynamic events.

The static events cover a professional presentation, a review of the engineering design, and a cost analysis for a total of 325/1000 possible points. A presentation is made to a panel of judges who evaluate the business case of the design, including business concepts like identification of market and profitability. In the design event, teams are expected to explain design choices and trade-offs between cost and performance and how well they integrated these concepts into the final design. Cost is constantly evaluated throughout the process, and each team must submit a Cost Report to enable the judges to assign a fair score.

The dynamic events test the actual performance of the car. First the car must pass a technical inspection to ensure safety and that all rules and parameters are fulfilled. The acceleration event takes place on a straight 75m course and simply measures the car's acceleration. The skid-pad event tests the car's cornering ability on a course of 2 constant radius circles. The autocross event is a half mile course with various turns and straights to evaluate maneuverability and handling without other cars on the course. Average speeds are expected to be 25 to 30 mph. Finally, the endurance and efficiency events are combined in an approximately 13.5 mile course of multiple laps with other cars. Again the handling and speed are tested while the length tests durability and reliability. At the conclusion of the course the fuel efficiency is measured and then scored.

Full CAD Model

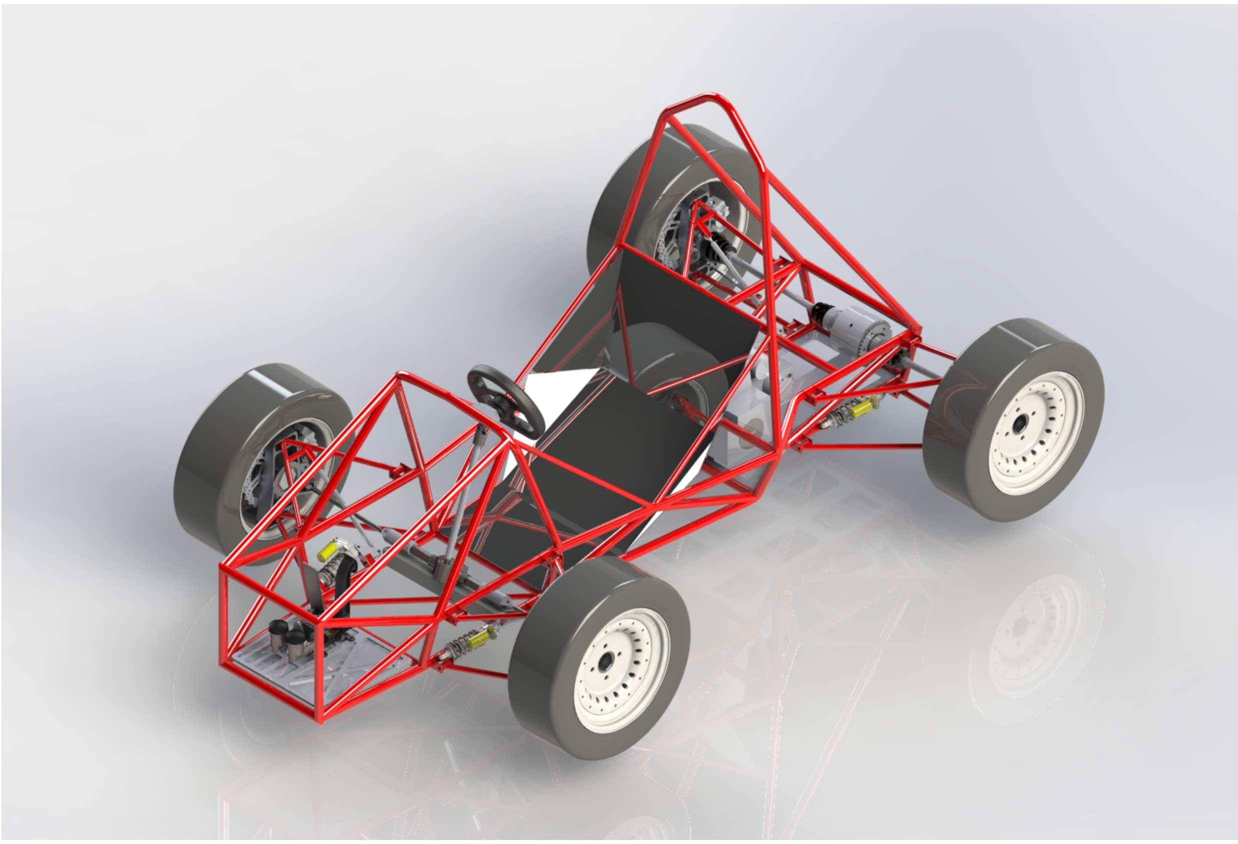


Figure 1: Full CAD Model of the Car

Drivetrain

Drivetrain Overview

For this project the drivetrain is defined as the system that transmits power from the engine to the wheels and all necessary components in between. Thus the major components of the drivetrain are an engine, transmission, differential, axles, and wheels. As the 2016 FSAE entry will be an entirely new car, every single one of these parts needed to be selected for a specific goal and properties and integrated with the other subsystems. Keeping in line with our goals for simplicity and affordability while remaining competitive, all available options for each element were researched to identify the best for the application. A single cylinder engine was selected for its simplicity, aiding maintenance and improving durability and reducing weight to bolster handling. A 2005 YFZ 450 with a big bore kit (500cc) was then purchased. The engine was then mounted to a constructed test stand to enable integration of the engine subsystems. An electronic fuel injection (EFI) system was also implemented to precisely control fuel input, improve efficiency, and conform to FSAE rules. The increased power needs of EFI required upgrading the stator to produce an additional 2 amps. A Honeywell MGT1238 turbocharger, generously donated by Honeywell, was added to the engine to boost the power. After many years of WPI's FSAE teams' challenges with a continuously variable transmission (CVT), a traditional manual transmission was selected. This would be natively integrated with the engine, cheaper, and significantly less work to tune and maintain while delivering similar performance. Based on the internal gear ratios of the engine and course speeds, a drive sprocket size was determined. A limited-slip clutch differential was chosen to improve handling during cornering by allowing different wheel speeds and application of power to the wheel with the most traction. A 1999 Honda TRX300 differential was selected and a housing was designed for its lubrication and mounting. Axles from the TRX300 ATV were selected and will be lengthened to fit the wheelbase. Finally, the wheels used by the 2012 FSAE car will be repurposed for this car as they are still in good condition and will enable the budget to be used on more critical areas.

Engine

Constraints

The official FSAE rules lay down some restrictions on the engine. According to rule IC1.1, the engine must be a piston engine using a four-stroke primary heat cycle with a maximum displacement of 610 cc. The airflow must also be limited by a circular restrictor with diameter of 20 mm for gasoline-fueled and 19 mm for E-85.

Research

The primary design decision in selecting an engine was the number of cylinders. More cylinders offer increased and better-balanced power at the cost of greater weight and complexity. The following provides brief descriptions of the available options and concludes with a weighted design matrix.

Single Cylinder

A single cylinder engine is the simplest available engine and a common choice for FSAE teams. It consists of a single large cylinder through which all the combustion occurs and power is generated. This simplicity offers several direct and indirect advantages. Single engine cylinder engines have fewer parts because they don't have multiple moving pistons that need to be precisely timed. This translates into weight savings – a typical single cylinder engine only weighs about 50 pounds- which enables better fuel efficiency, handling, and acceleration. Only having one cylinder greatly simplifies the design and reduces the number of parts needed which leads to greater durability and reliability. However these engines only produce about 50 hp. The large explosion needed by the single cylinder creates vibrations and reduces the comfort of the ride and may require additional mounting or dampening. Additionally used single cylinder engines are more expensive to purchase due to supply, although the reduced maintenance and part costs help to reduce this gap.

Twin Cylinder

A twin cylinder is a compromise between the single and four cylinder engines. By adding another cylinder a greater amount of power is gained, improving output to 65-80 hp. This comes at the cost of an additional 20 pounds of weight and added complexity to the intake, exhaust, and other systems. Twin engines are also less available in the used engine marketplace and tend to cost more than the other options. Additionally very few FSAE teams use twin engines, preferring the single or four cylinder options.

Four Cylinder

The four cylinder engine offers the greatest amount of power at the cost of complexity and weight. They are able to achieve over 100 hp, although the restrictor limits maximum performance. A four cylinder engine and the necessary elements tend to weigh about 50 pounds more than a single cylinder, increasing the estimated weight of the car by 10%. By splitting the power generation across 4 smaller cylinders, a much smoother ride is produced which is important to the goal of rider comfort. The cost of used four cylinder engines is also lower than the other types as their original vehicles tend to suffer damage and be reduced to salvage. These savings are reduced by the added complexity in delivering air and fuel and maintaining piston synchronization. A greater number of moving parts increases the opportunity for something to go wrong.

Design Matrix

		Single Cylinder		Twin		Four Cylinder	
Decision Factor	Weight	Score	Value	Score	Value	Score	Value
Simplicity	10	10	100	7	70	6	60
Reliability	8	8	64	6	48	4	32
Cost	7	8	56	4	28	8	56
Power	8	7	56	8	64	8	64
Weight	6	9	54	7	42	6	36
Weighted Score			330		252		248

Table 1 Engine design matrix

The design matrix shows the single cylinder engine to be the best choice for this application, primarily for the relative simplicity. It is better to have a less advanced but still functional car. Cars are complicated and difficult to maintain; even at the FSAE competition many teams fail events due to engine difficulty and other technical issues. By selecting a simpler design the engine and car will have a better chance of being able to perform and last through all of the events. Although a single cylinder engine offers less power, this downside is limited by the reduced weight that enables greater acceleration and handling. These properties are significantly more important than top speed in order to score well in the events of the FSAE competition. Finally, the lower power can be mitigated by adding a turbocharger to remain competitive without drastically increasing complexity.

Selecting Engine Model

With a single cylinder engine option selected, the next step was to source an engine. The team reached out to several companies and local motorsports businesses seeking sponsorship through a donated engine but to no avail. Thus the MQP group had freedom to select the engine model to be used. We decided it would be most economical to purchase the engine through eBay, protecting ourselves by carefully reviewing the seller’s reputation and ensuring a great return policy should some issue occur.

Make	Model	Origin	Valid Years (Electric Start)
Honda	CRF450R	Dirt Bike	All (from 2004)
Yamaha	WR450F	Dirt Bike	All (from 2003)
Kawasaki	KLX450	Dirt Bike	From 2008
Honda	TRX450ER	ATV	From 2006
Yamaha	YFZ450	ATV	All
Kawasaki	KFX 450R	ATV	All (from 2008)
Suzuki	LTR450	ATV	All
KTM	450 EXC/XC-F	Dirt Bike	From 2006 (at least)
KTM	500 XC-F	Dirt Bike	Seem to have them

Table 2 Single cylinder engines considered for purchase

A Yamaha 2005 YFZ 450 single cylinder engine was purchased. The engine was barely used, recently rebuilt, and outfitted with a big bore kit, upgraded cams, and a stoker kit increasing the displacement to about 500cc, a 10% increase. The owner had been looking to build a competition ATV and procured the engine but never got around to completing his project. Visual inspection of the engine confirmed it to be in good shape and nearly complete with only minor hoses and pipes missing. The engine achieved compression; the next step was to get it mounted on a test stand and purchase the remaining subsystems.

Figure 2: YFZ 450 Engine

To enable rapid integration of the engine with the CAD model, the Autodesk 123D Catch iPhone app was used to capture a model of engine. By taking 24 pictures, the free software was able to build a rough model that could be used to estimate packaging before a more detailed version could be produced. By scaling and cleaning the .stl model in the Meshmixer program provided for free by Autodesk, an accurate .stl file was produced for use in SolidWorks. While sufficient for a rough draft, a more detailed model was needed for accurate mounting and ensure compliance with the frame. Thus the major features and critical dimensions were used to produce a more detailed CAD model.

Figure 3: Autodesk 123D Catch iPhone App CAD Model

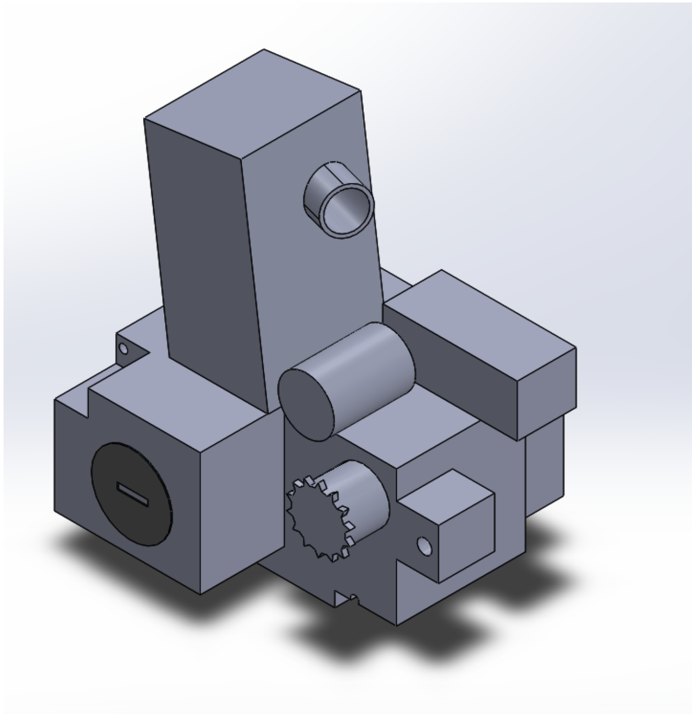


Figure 4: YFZ450 CAD Model

Test Stand

In order to confirm that the engine is working and obtain important measurements and integration with other subsystems before the frame's completion, a test stand was needed. There was an engine stand available from a prior MQP that could be modified to mount the new engine. Thus, the necessary modifications were made and the engine successfully mounted.

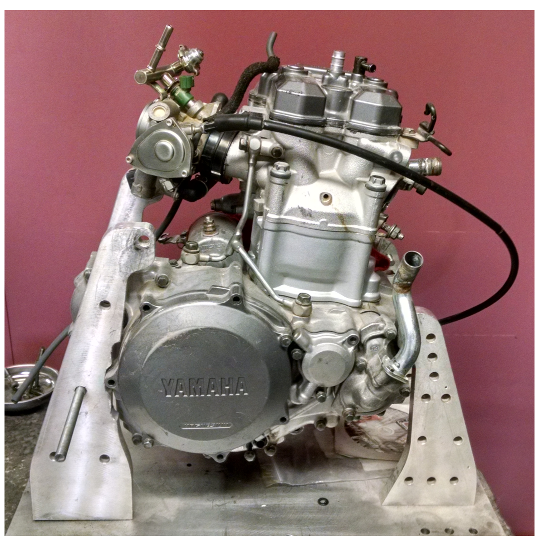


Figure 5 Engine Mounted on the Test Stand

Accessories

The purchased YFZ450 contained a complete engine but lacked many accessories needed to attain full operation. This was by design; the engine was an excellent price and the cost of the related accessories was not prohibitive. In addition, some of the stock parts would have needed replacement due to modifications being made; a carburetor would not have been used because the engine will be fuel injected. The following were sourced online and purchased: oil tank, oil breather tank, exhaust pipe, throttle body, and muffler. The oil tank and oil breather tank were both from the original YFZ450 and did not need additional modifications. A stock exhaust from a 2010 YFZ450 was selected because it was still able to fit the outlet and came packaged with the muffler. The stock muffler is large however and can be shrunk for better packaging. One advantage of selecting the YFZ450 was that while the 2005 model was not EFI, from 2007 on it was only produced as EFI with minor changes. Therefore a throttle body from the EFI version was purchased and able to fit onto the engine without any adaptors. In addition to the large purchases, smaller components like gaskets, bolts, and clips will need to be purchased as required.

Figure 6: YFZ450 Muffler

Improvements and Modifications

While an excellent base, the purchased engine was not complete. There are 2 subsystems of the engine that need to be modified: the stator and fuel supply system. As mentioned earlier, the stator will only generate 120W at 14V, or about 8.5 amps. This is insufficient power to run the ECU, fans, and EFI system. However this is an easy fix as there are many companies that offer custom windings. We located several that are able to increase the power output by 2A. Customrewind.com is able to provide the service for the cheapest. The fuel supply system is less simple; it was converted from a mechanical carburetor to electronic fuel injection (EFI).

Electronic Fuel Injection

The 2005 model of the YFZ 450 is carbureted and was upgraded to EFI for several reasons. EFI offers greater control of fuel being supplied to the engine which increases efficiency and performance. The turbocharger will also affect the amount of fuel needed and the greater control will enable precise tuning for maximum power gains. As the car already has an advanced ECU, the majority of the conversion work needed was mechanical by adapting old parts and selecting new components.

The timing of the fuel system was determined by a raised tooth pattern on the flywheel that was then detected by a VR (variable reluctance) sensor. The pattern consisted of a single long tooth which was adequate for the carburetor. However for better performance the exact position of the flywheel and by extension, cylinder, needed to be known more frequently than once every rotation. To this end, a slotted 12-1 missing tooth pattern was machined into the flywheel so the precise location is better known. 11 slots were machined into the flywheel at equal intervals (30°) calculated on 12 slots. The missing slot is used to mark a full rotation and offer a frame of reference to the ECU. The placement of the missing tooth was critical. Reaching out to Haltech led it to be placed 70-80° after TDC. However, the VR sensor that detects the missing tooth is 90° from TDC and factored into the placement.

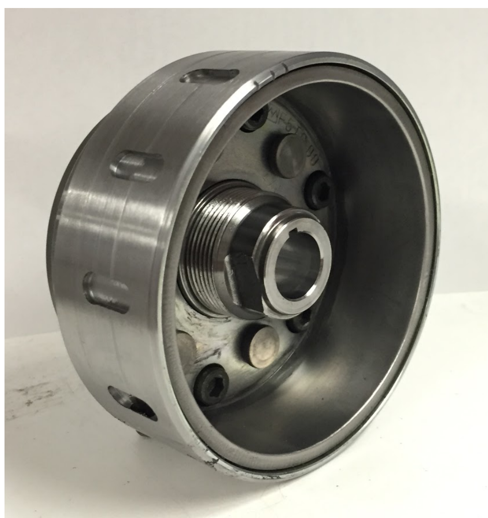


Figure 7: Machined 12-1 Flywheel

Transmission

The two options for a transmission were either persisting with a continuously variable transmission (CVT) or the traditional manual transmission built into the engine. A CVT consists of 2 cones with a sturdy belt between them. As the belt slides up and down the cones the diameters change, thus altering the gear ratio in a steady fashion. This offers many advantages: increased fuel efficiency, eliminating the jerk and lag of switching gears, customizable gear ratios, and faster acceleration. However in order to perform better, the CVT must be tuned and maintained which is a very challenging task. WPI’s FSAE teams have attempted a CVT since

2009 and never really found success while dedicating large amounts of time to maintaining and fixing it. As most engines come with a manual transmission, a CVT would be an added feature that complicates packaging and stretches the available engineering resources.

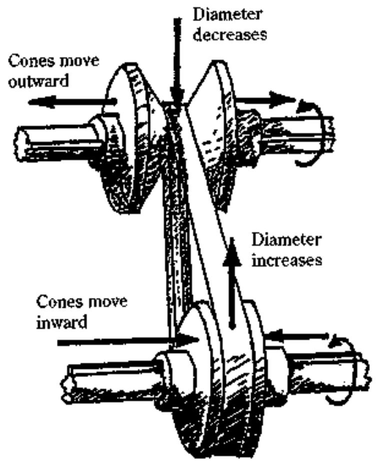


Figure 8: Diagram of a CVT

In contrast, a manual transmission is the typical 5-speed clutch-operated version that many are familiar with and it requires less set-up and maintenance. Additionally most dirt bikes and ATVs that we considered sourcing an engine from were manual, enabling an all-in-one solution that simplifies packaging and design. The YFZ 450 was no exception. The main reason for selecting a manual transmission however was to fulfill the project goals of simplicity and durability, freeing the time and resources of both this team and the next to ensure the car is functional and deal with other problems that will crop up.

Turbocharger

One disadvantage of the single cylinder is that is produces a relatively low amount of power at about 50 hp. This number can be boosted about 15% by making use of the waste exhaust pressure through use of a turbocharger.

There are 2 types of "chargers"- a supercharger and turbocharger. Both improve the engine's output through the same mechanism: compressing the air sent to the engine. By compressing the air, more can fit into the cylinder and more fuel can be supplied. A bigger explosion can generate more power and thus increase the power supplied by the engine. The mechanisms used to compress the air differ though. A supercharger uses a belt attached to the crankshaft to compress the air, producing a net gain. A turbocharger uses the pressure of the exhaust to drive a turbine that compresses fresh air before entering the engine. A turbocharger achieves the best gains when the engine is already operating at high levels whereas a supercharger operates optimally at low speeds. As the average speed of the FSAE events is 25-35 mph and focus heavily on acceleration and handling, a supercharger would be the preferable solution to deliver a power boost where needed most. Unfortunately superchargers are expensive, starting at around $2000 and infrequently used for this application. A turbocharger can still provide excellent and more efficient gains as it takes advantage of exhaust instead of taking power from the output shaft. However they do suffer from lag as it takes a little bit of time for the turbo to make use of the increased amount of exhaust. Additionally turbochargers have been used before in FSAE and tend to provide gains of 15%.

A Garrett turbo was found in the SAE room and made available for use, but discovered that Honeywell offers to sponsor an FSAE a turbo if it can be proven to be a good match for the engine. The following are calculations used to show that the MGT1238 is a good match for our engine and targets.

Equations:

Wa = HP*A/F*BSFC/60
MAP = Wa*R*(460 + Tm)/(VE * N/2 *Vd)
P2c = MAP + Ploss
P1c = MAP – Ploss
Πc = P2c/P1c

Airflow Actual (lb/min)	Wa	6.85
Horsepower target	HP	65
Air/Fuel Ratio	A/F	11.5
Brake Specific Fuel Consumption	BSFC	0.55
Manifold Absolute Pressure (psia)	MAP	23.06
Gas Constant	R	639.6
Intake Manifold Temp (F)	Tm	150
Volumetric Efficiency	VE	0.95
Engine speed (RPM) @ peak power	N	8000
Engine displacement (c in)	Vd	30.5
	Boost Pressure	8.36

	Atmosphere Press	14.7
Compressor discharge pressure (psia)	P2c	23.06
Pressure loss b/t compressor and manifold	Ploss	1.5
Compressor inlet pressure	P1c	13.2
Pressure ratio	πc	1.99
	MAP (Kg/s)	0.0518

Table 3: Calculations used to locate position on MGT1238 turbo map

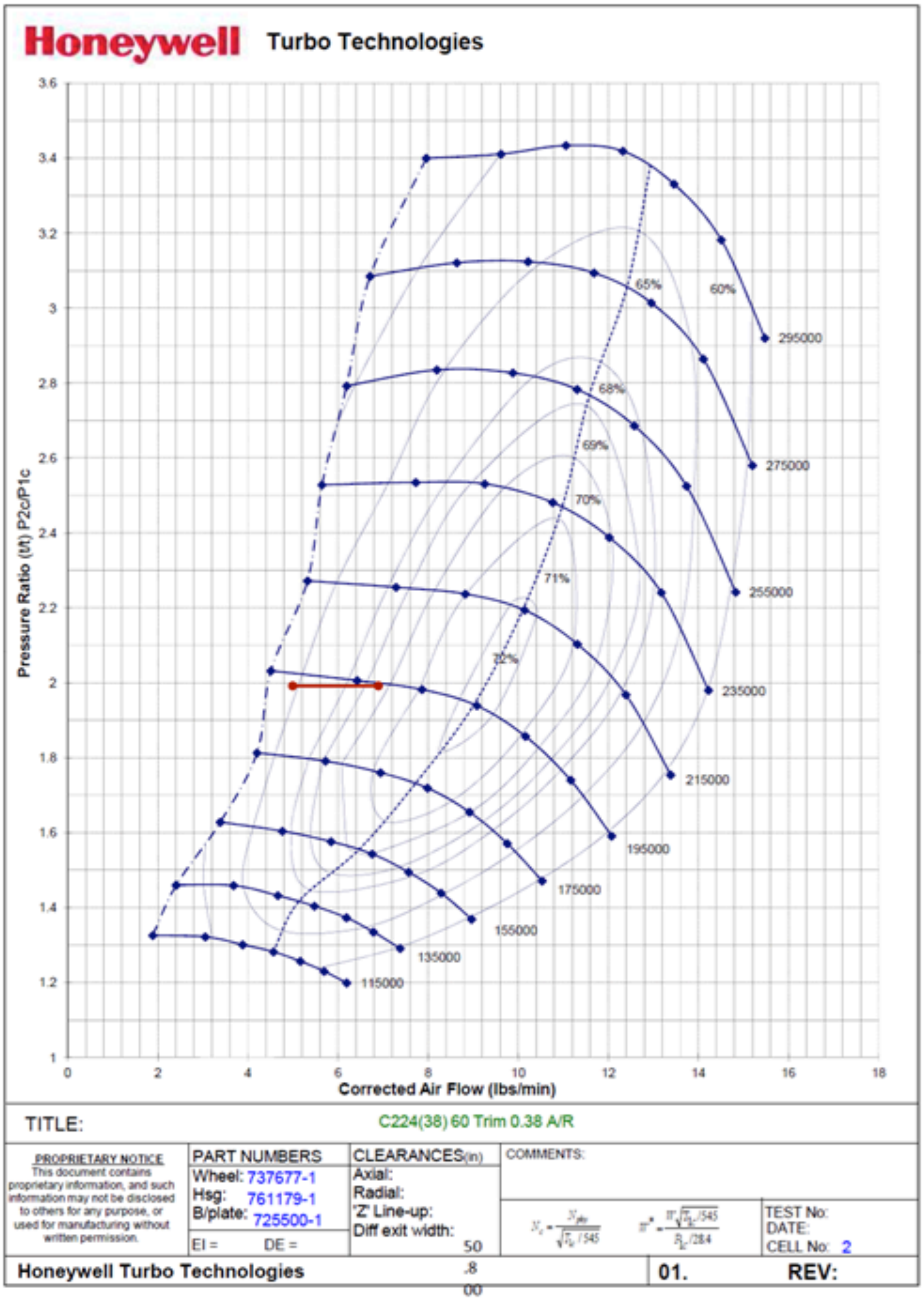


Figure 9: MGT1238 Turbo Map

Fuel Choice

Formula SAE rules state that vehicles at competition must be able to operate off the fuels provided. This limits selection to octane 91/93, 97/100, and E85. The octane fuels are derived from petroleum and as rating increases, so does the difficulty in premature ignition occurring that can cause severe damage to the engine and fuel lines. E85 is produced from an 85% ethanol and 15% gasoline mix and has several important properties. Its equivalent octane rating is about 105, meaning it is even less prone to knocking. As an alcohol, it is capable of drawing more air into the fuel/air mixture which both provides more power and cooling. The cooling is especially important with the added turbocharger: a combination of this effect and the small size of the turbo makes an intercooler unnecessary. This will reduce cost and increase simplicity by virtue of having fewer components. For these reasons, E85 is the preferred fuel choice. However, E85 does have 2 disadvantages. It is less fuel-efficient because it is less energy dense than gasoline, which will affect scoring. It can also be difficult to source and more challenging store than regular high-octane gasoline. Fortunately many gas stations in Worcester offer E85, including the Park Avenue Mobil station. It does draw water more easily though and requires careful storage that must be considered.

Differential

A differential enables the wheels to spin at different speeds, delivering power where there is the most traction and significantly improving handling. In order to determine what differential would be most suitable to the project, research was done to several common differentials and several specific models for the types.

An open differential is the simplest of all differential types. Although it allows the wheels to spin at different speeds, it applies the same amount of torque to both wheels. This limits the torque delivered to the wheels when one loses traction as the torque is split between the 2 but only used by 1. Open differentials are the least expensive and easiest to integrate due to their simplicity. Turning and handling is a substantial feature of the FSAE events, however the simplest solution cannot be selected and it must achieve better performance, especially in adverse conditions.

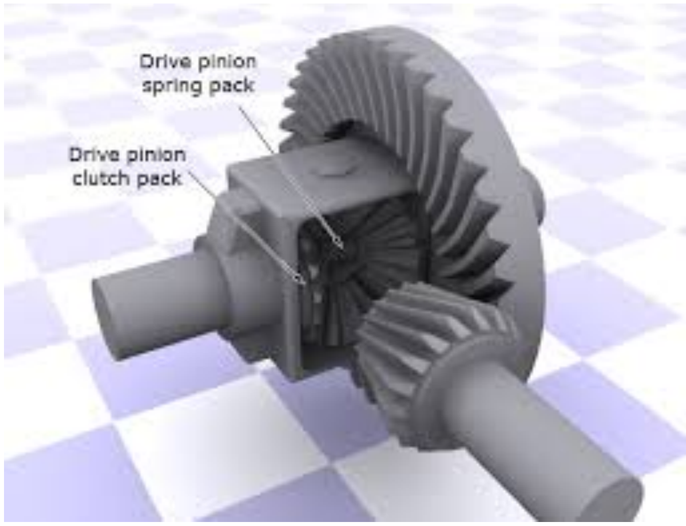


Figure 10: An Open Differential

A limited-slip differential (LSD) uses a variety of mechanisms to ensure that as one wheel loses traction, the other wheel receives more torque to make up the difference. This substantially improves handling, particularly in adverse conditions like rain. However, they are significantly more expensive and are subject to maintenance and wear. There are a variety of LSDs available that use different mechanisms to produce the desired result.

A clutch-type LSD uses a spring pack and set of clutches to adjust the torque delivered to specific wheels as the speed ratio changes. It can be tuned by adjusting the springs and clutches to create the desired performance. This can help mitigate the body roll effect. As a car is turning the outside wheel will attempt to rotate faster which then increases the amount of torque sent to the inside wheel. This creates slippage and the opportunity to induce understeer. Additionally the clutches and springs will need periodic maintenance and the clutches can wear out with time. However, the clutches are predictable and will engage reliably without internal slippage which improves performance.

A Torsen differential is a type of LSD that uses gears to create a built-in ratio called the torque bias ratio (TBR) that determines the amount of torque sent to the wheel with better traction. It is more durable because it doesn’t contain clutches or other complicated mechanisms. However it is vulnerable to acting like an open differential if one wheel completely loses traction. It also takes time to engage as there is slippage in the gears.

A 1999 TRX300 Fourtrax clutch LSD was located in the SAE room and selected for use in the car. As a LSD, the handling will be improved, especially in catastrophic conditions like total traction loss. Unfortunately, only the gearing was found- it lacks a housing to contain the lubrication oil and provide mounting points. Additionally it is driven by a pinion gear whereas this project will be driven by a chain and sprocket. This is complicated by the fact that much of the differential is made from hardened steel and is extremely difficult to machine. Therefore the ring gear was entirely removed and a new cap purchased to avoid machining the precise geometry that interfaces with the axles.

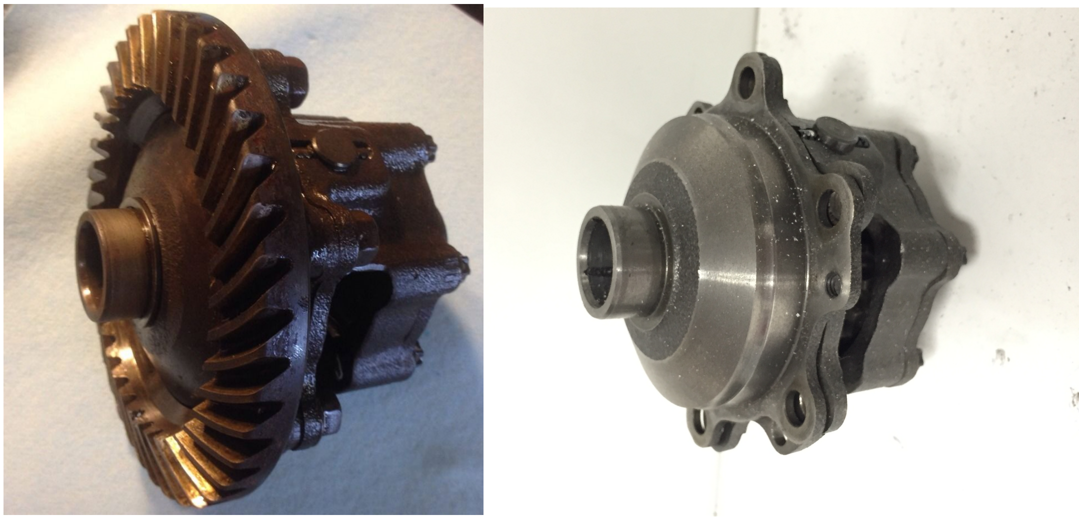


Figure 11: The TRX 300 Differential Before and After Ring Gear Removal

Drive Sprocket

A driven sprocket was selected to transmit power from the engine to the rear axle. Unmodified, the engine's output goes to a 520-chain 14 tooth sprocket. Selecting to use a driven sprocket allowed simplicity of design and flexibility in engine placement. The size of the driven sprocket was an important design point as it determines the speed for each gear. The following chart lists the internal gear ratios and output of the manual transmission.

Gear	Ratio	Decimal	Total
Primary Ratio	62/22	2.818	
Secondary Ratio	38/14	2.714	
1st	29/12	2.416	6.80
2nd	27/14	1.928	5.43
3rd	25/16	1.562	4.40
4th	23/18	1.277	3.60
5th	21/20	1.05	2.96

Table 4: Gear Ratios of YFZ 450

With data on the engine's gear ratios located, the next step was determining the speeds at which the engine would be operating in what gear. A study of the 2012 FSAE competition was performed that compiled the average speeds for the acceleration, autocross, and endurance events for the best, 5th best, and 10th best time. The gear ratio is important; if too low then the car will not be able to reach the speeds necessary to perform and if too high then optimal power will not be achieved. The following table summarizes the events from the study of FSAE teams at the 2012 competition.

Event (2012)	Dist (mi)	Best Time (s)	Avg. Speed (mph)	5th Best Time	Avg. Speed (mph)	10th Best Time	Avg. Speed (mph)
Acceleration	0.0466	4.11	40.81	4.276	39.23	4.33	38.74
Autocross	0.5	51.6	34.88	53.9	33.40	56.66	31.77
Endurance	13.66	1067	46.08	1153	42.65	1213	40.54

Table 5: Average Speeds of 2012 FSAE Placers

Based on the acceleration event's average speed of high 30s, the top speed is greater than 70 mph. Autocross is much slower as the course contains many turns and obstacles and has an average speed of about 33 mph. The endurance is faster, pushing into the 40s. A calculator was created in a spreadsheet that took in the number of teeth of a sprocket and produced the final gear ratio and speed at peak power. By looking at available sprockets and testing gear ratios, a 34-tooth sprocket was selected. It is capable of achieving 70+ mph and fits into the autocross and endurance speeds well without necessitating excessive gear shifting. The calculations are based on 20.5 in tires at 8500 RPM producing peak power; this number may vary from actual engine performance and the effect of the turbocharger. It is recommended that the calculations are performed again once more data is collected

Speed = RPM*Final Reduction*(Wheel Size*3.1415)/12/5280*60sec/min

Gear	Reduction	Final	Speed (mph)
1st	6.81	16.53	31.35
2nd	5.43	13.19	39.29
3rd	4.40	10.69	48.49
4th	3.60	8.74	59.31
5th	2.96	7.19	72.14

Table 6: Speeds at Peak Power for 34-Tooth Driven Sprocket

Axles

As the differential is sourced from a TRX300, it made sense to select the associated axles with the proper spline to avoid extra costs and unnecessary effort. Two left TRX300 ATV axles were purchased because they are longer than the right-sided version. While the splines integrate with the differential, the stock length is not sufficient to reach the uprights and hubs and will need to be lengthened.

The alternative to modifying stock axles would be purchasing custom axles from a professional company like Taylor racing. However, the cost would be at least double and closer to triple the cost of modification for similar results. Therefore the TRX300 axles were purchased and are available for modification once the final dimensions are determined. Approximately 3-4 inches will need to be added to each axle. The exact dimension is still subject to modification based on adjustments to the suspension arms and final engine placement and differential. Regardless, the procedure remains the same. Chromoly steel tubing of equal diameter to the axle center will be purchased and cut to the required length. The axle will be cut in the center with the tubing placed between the halves and sealed by welding. This will result in axles of the proper length and of sufficient strength to maintain its integrity when subjected to testing conditions.

Differential Housing

A differential contains many precise parts and requires both protection and low friction to operate. The TRX300 differential's stock housing was not able to be used after the adaptation to a drive sprocket. Therefore a new housing was designed to protect the delicate internals from particles and debris and contain lubrication fluid. The housing must also contain an interface for mounting to the frame. With these considerations, a lightweight aluminum housing was designed to fulfill these requirements. The design consists of 2 cylindrical aluminum tubes that the differential is mounted to using the 6 existing mounting holes within the differential and contained between 2 plates.

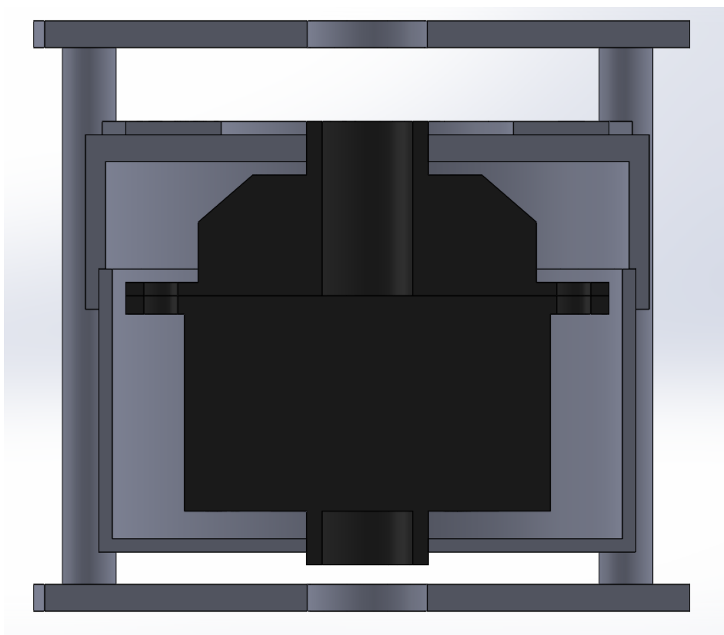


Figure 12: Differential Housing Cross Section

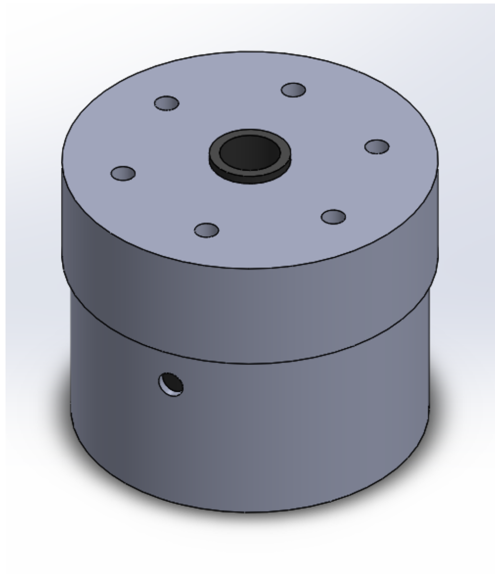


Figure 13: Differential Housing

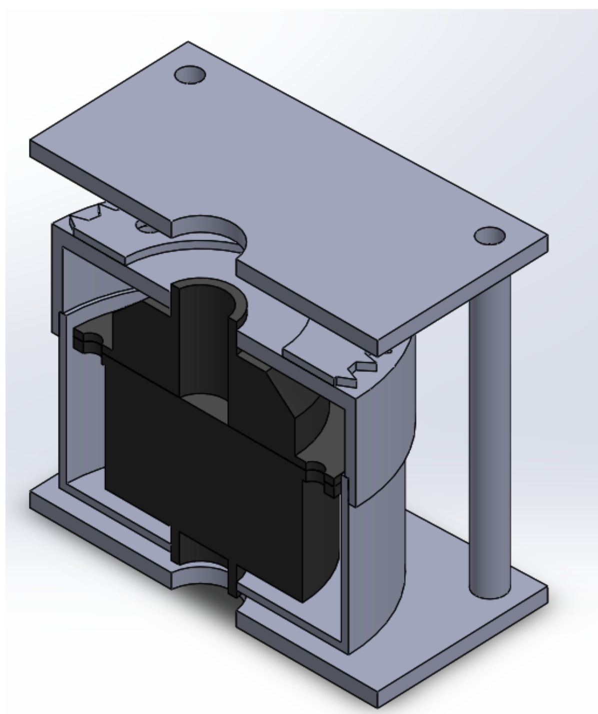


Figure 14: Cross Section of Housing with Sprocket and Mounting Plates

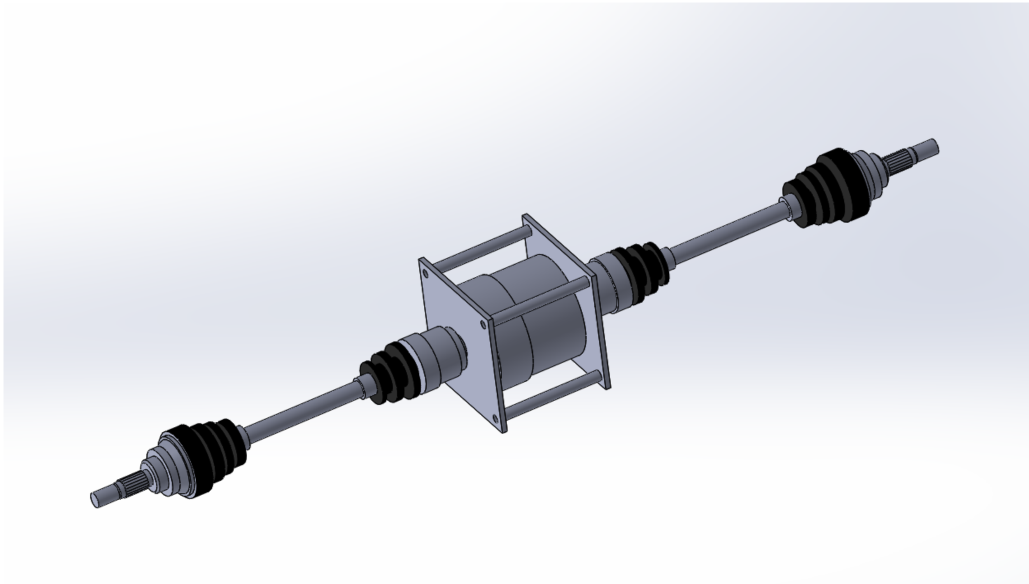


Figure 15: Differential with Axles

Wheels

The same 13 inch Hoosier 43128 FSAE Slick tires as previous FSAE teams. Our team had familiarity with these tires and size, allowing benefits and enhancements of the frame from last year's project to be recycled into this edition. The tires are also in good shape and avoiding unnecessary purchases enables greater improvements in the rest of the car.

Frame and Suspension

Frame Overview

In the design of a racecar, torsional rigidity is critical aspect in frame design. Any torsional flex in the frame is equivalent to adding another spring to the system, making suspension tuning unpredictable. As with any structure made out of materials that deflect it is nearly unavoidable to have some chassis flex during hard cornering. However, the goal during the design of a frame is to minimize deflection to allow for consistent suspension performance with immediate and predictable results. Therefore, in the design of a frame the goal is to make the frame as stiff as possible. Torsional rigidity is more important this year due to the use of an independently acting suspension vs the 2012 car's original swing-axle rear suspension. With an independent suspension, the roll stiffness is important to keep suspension kinematics intact. Another key constraint in the design of the new frame is minimizing weight. During the design of the new frame all tube sizes that were not explicitly dictated by the Formula SAE rulebook have been analyzed under loading conditions. Using the results from FEA simulations the tubes were reduced in size until they would be strong enough to support the loads while still offering a weight advantage. The new frame must weigh less than the current frame while providing better overall torsional stiffness. The frame also has a large impact on the center of gravity and weight distribution of the car. To maintain the best center of gravity and weight distribution the frame has been stretched slightly to accommodate taller drivers and the roll hoop was raised to help the driver sit more upright in the car. This helps improve the responsive handling characteristics of the car during dynamic events. The design templates dictated by the Formula SAE rulebook provided several of the constraints for designing the frame. The design of the 2012-2014 Formula SAE car also provided a number of constraints to base the new design on. A study of the 2012 frame was conducted to identify areas that needed to be address while designing the new frame. Alterations were made to the seating position, main hoop supports, front bulk head, and rear sections. Due to budget and time constraints, only chromoly tube space frames were considered during the design and research of this frame. To save time the frame was manufactured by VR3 Cartesian Tubing who is very well known within the FSAE community for very high quality manufacturing and accurate fitment. To stay within the project budget the frame will be fixtured and welded in-house.

Study of the 2012 Frame

The first step in the design of the new frame was to take an in-depth study of the current frame that was designed for the 2012 car. This car originally used a solid rear swing axle. The car was converted over to an independent rear suspension during the 2013 MQP. One of the main design goals for this frame was to improve the overall size of the frame to allow it to fit larger drivers. This worked well and was a great improvement over the past car. However, during the conversion to an independent rear suspension the seating position was moved forwards. This negated the addition frame length. Next the foot well area was designed with the CVT transmission in mind so the pedal area is not large enough for the addition of a clutch pedal for a manual transmission. Several ergonomic problems exist with the current driver cell. The roll hoop support bars caused issues while entering and exiting the car. The front roll hoop impacts taller driver's knees. Issues also exist with the 2012 frames side impact bars. The side impact bars are tilted upwards towards the rear of the frame. This makes it hard to meet the rule requirements for side impact protection. Another rule issue exists with the main roll hoop. Due to the near 0 degree inclination, the hoops support bars requirements can change with small changes in ride height. This could cause issues during technical inspection.

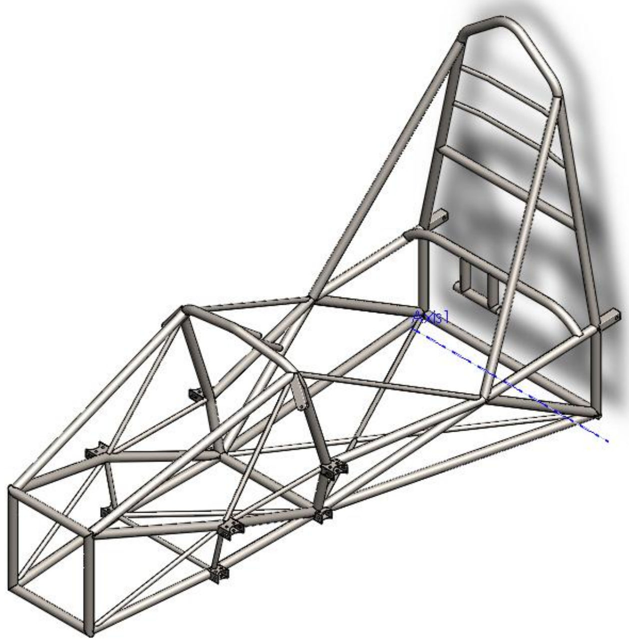


Figure 16 2012 FSAE Frame

Objective

To design a new frame that would address the known issues in the 2012-2014 Formula SAE car while still meeting the Formula SAE rules the following needs to be done.

- Any alterations to the frame must meet Formula SAE rules.
- Meet all manufacturing requirements set by Cartesian Tubing
- Changes to the driver's seating position must be accounted for to increase overall legroom, foot room, elbow room, and headroom. This is to allow for a greater range of physical driver sizes.
- Decrease overall frame weight as much as possible while still maintaining structural rigidity. The overall weight goal for the 2015 frame is less than 75 lbs.
- Provide appropriate mounting points for suspension and engine.
- Maintain suspension geometry for the front and rear.
- Maintain maximum serviceability while keeping packaging constraints in mind.

Research

After a review was completed of past years frame several additional publically available FSAE frames were researched to get a better understanding of different FSAE frame layouts. Due to rule limitations and size constraints it is typical for all FSAE frame share similar aspects to each other. The biggest constraint during the design of the frame is the manufacturing abilities of Cartesian tubing. Several example frames were studied. The first was an MIT frame that was hosted by the team. This frame features the use of square tubing on the lower rails for easier mounting of components. It also features large radius roll hoop. The triangulation on the front of the frame features a single diagonal and a split side impact diagonal. The upper side impact support bars only support down to the lower front hoop. The rear main hoop support bars are triangulated by two sections.

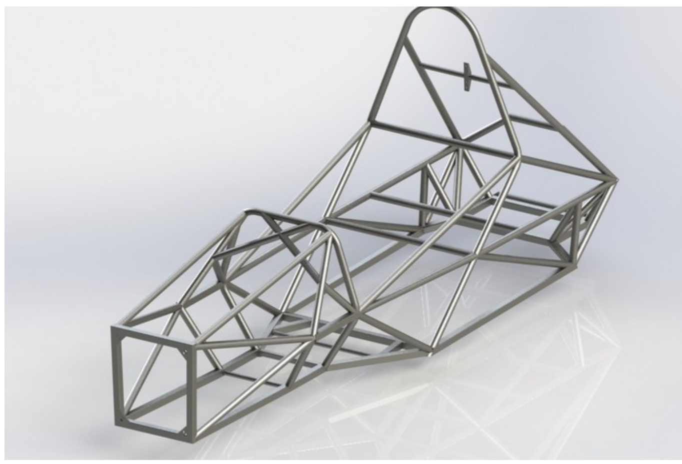


Figure 17 MIT FSAE Frame

The next frame studied was from the University of Dalhousie. This frame uses similar aspects from the MIT frame such as the use of square tubing in the base of the frame and similar triangulation methods. The harness bar is moved rearwards in this design due to the driver sitting under the main hoop.

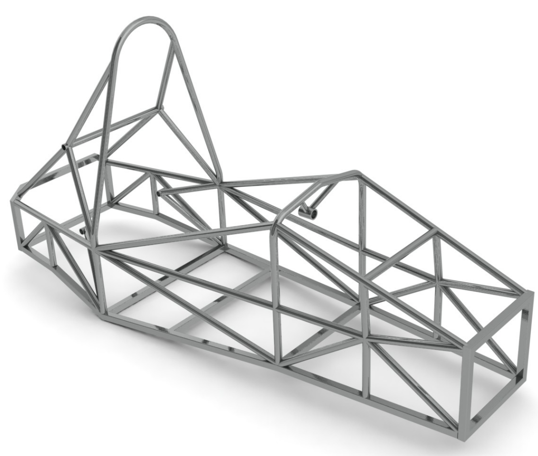


Figure 18 Dalhousie University

The next frame studied was produced by Cartesian as an example frame to get teams started. Using a typical FSAE layout the frame is designed to be modified by teams for use as a starting point. This is important since Cartesian has designed the file with setup 3D sketches and weldment profiles for use with their manufacturing processes. This provided the easiest and simplest way to move forwards with the new frame's design. This frame was selected as a base for the new cars frame. All geometry will be restructured to meet the performance specifications outlined.

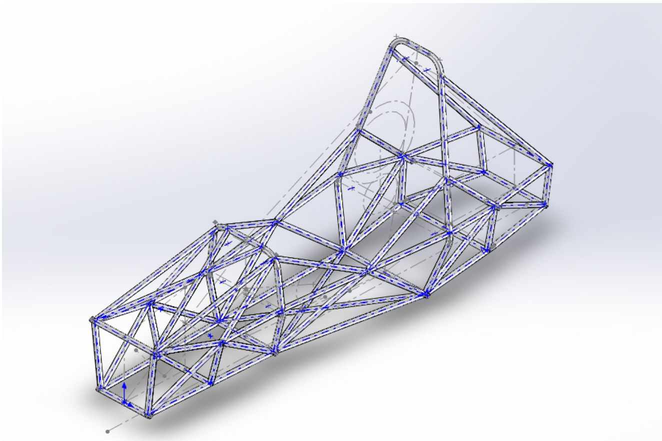


Figure 19 Cartesian Tubing Frame

Design

Preliminary Design Concepts

Starting with the Cartesian frame, two-concept frames were created. While the overall concept of the frame was already decided upon there were a few small design details that needed to be addressed. Each of these frames features a similar roll hoop setups to the 2012 car due to its simplicity and optimization. The first concept uses an angled frame bottom to help lower the center of gravity of the car while the second frame ops for a flat bottom to help with manufacturing and setup.

Concept 1: Angled Lower Frame Rail

The first frame concept uses an angled lower frame rail to help push the overall center of gravity of the car as close as possible to the ground. This helps lower the driver by around one inch. However, it comes at a price. Due to the angles involved with setting up the frame, it is difficult to fixture correctly. It also adds additional off axis bends to the frame, which does not help improve the overall stiffness.

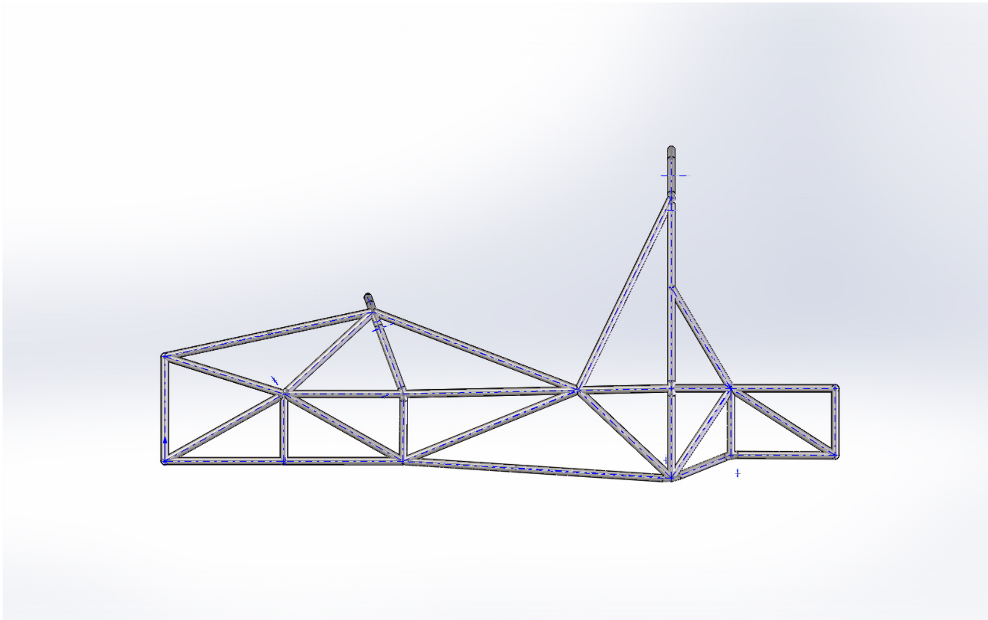


Figure 20 Angled Lower Frame Rail

Concept 2: Flat Frame

The second method is to make the whole bottom of the frame flat. This has several advantages; first the flat underside helps greatly with fixturing and welding setup. This is due to the ability to lay the frame on a flat surface during welding setup. Next, it uses less off axis bends than the angled method.

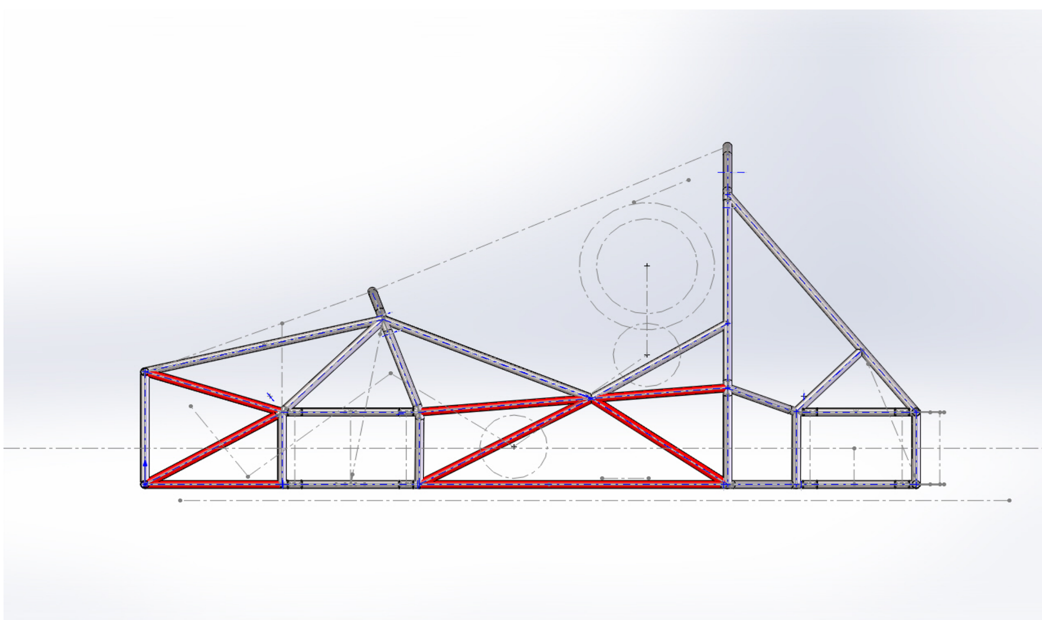


Figure 21 Flat Frame

Concept Selection

Due to ease of manufacturing and simplicity of the flat frame it was chosen for the final design. As stated before the frame is greatly dictated by the FSAE rules, packaging, and ergonomics. As such its design is more driven than created. The chosen design will allow for easier welding and set up than previous years, this will result in better tolerances during manufacturing and a higher quality final product.

Final Design 1st Iteration

Once a general set of dimensions were decided upon the Cartesian frame model was modified. The structure of the model was completely reworked to be easier to work with. After the general frame dimensions were entered, the suspension pickups were defined. To strengthen the frame triangulated sections were created in the base and side box sections. To speed up the design process the 2012 frame was used as a reference to the placements of these triangulations since the design was well optimized by the previous team. Also the old frame was used as a reference to the placement of the harness, hoops, and suspension pickups. Frame tubes not required by the Formula SAE rules were then analyzed independently for weight savings. After adjusting all of the appropriate tube sizes, final adjustments were made to the overall dimensions and geometry of the frame. Careful attention was given to fitting the 95th percentile male and 5th percentile female (called PERCY). PERCY is a template used during the technical inspection. The goal is to allow PERCY to sit as upright as possible while still keeping the roll hoops within a reasonable size to pass the 2" helmet clearance requirement. A 3D person model was also used to help with width requirements in 3D space.

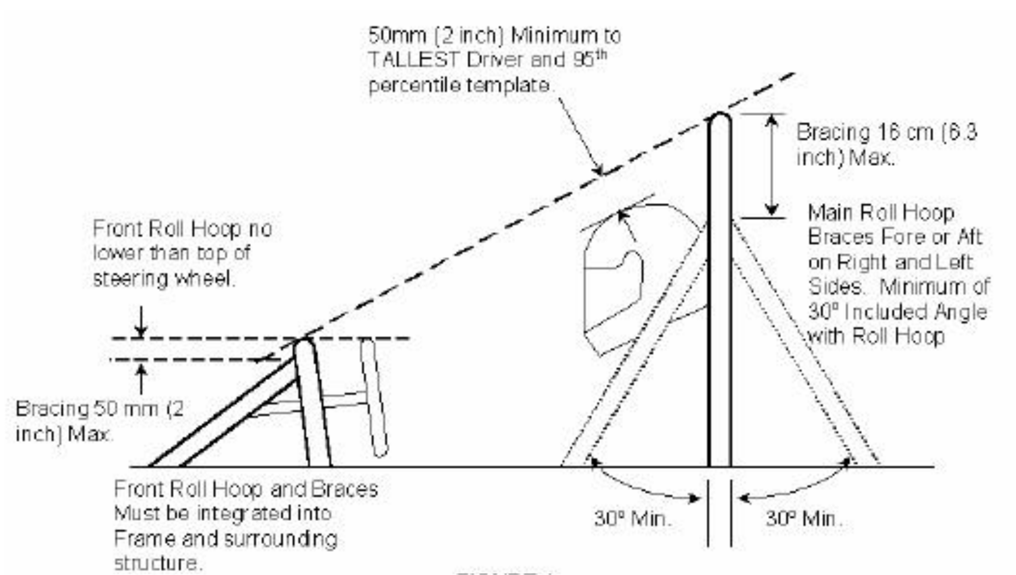


Figure 22 Roll Hoop and Helmet Rules

Above is shown the helmet clearance requirements, the driver's helmet should have at least two inches of clearance between the plane created by the tops of the main and front roll hoops, indicated by the dotted line in Figure 20. The main and front roll hoops were adjusted to accommodate the higher position due to the drivers more upright seating position. The change in the seating position also caused the head restraint supports and the harness bar to need adjustments. The last major alteration was to the foot well. The previous car did not have enough foot well width to fit an extra pedal to operate the clutch on the new engine. To fix this the front suspension pickup points were widened to provide more room. Also the front bulk head was flipped sideways to widen the front section of the frame.

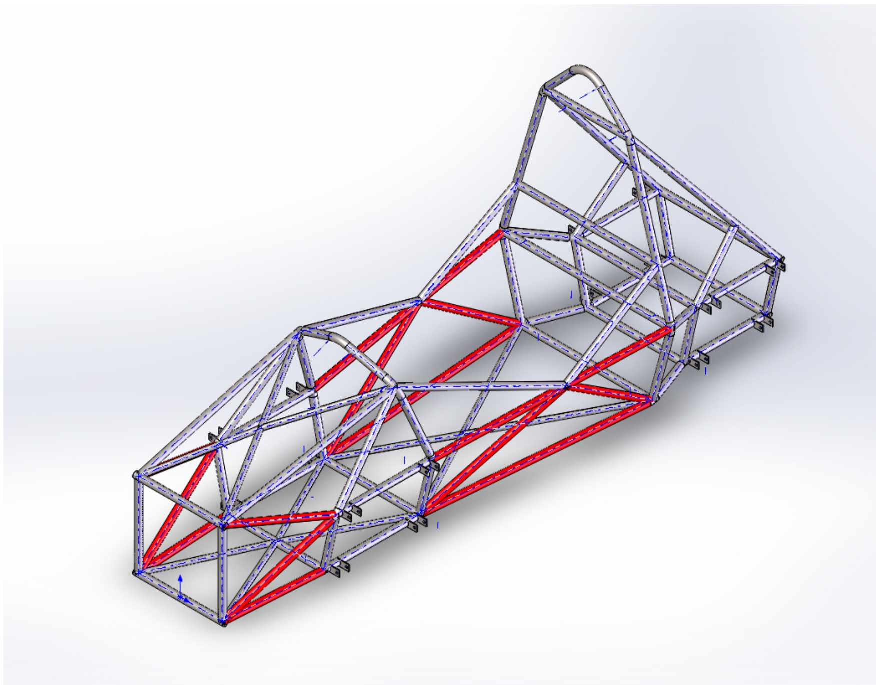


Figure 23 Final Frame

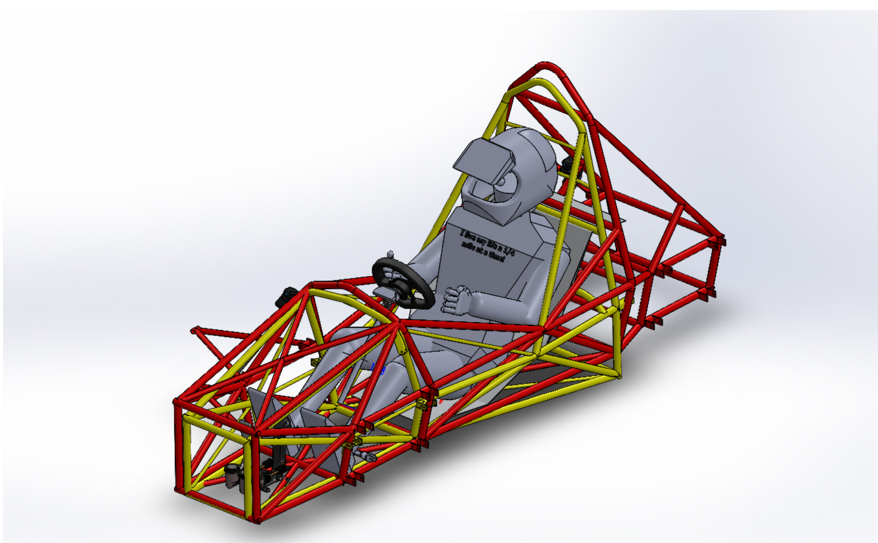


Figure 24 Comparison with old frame

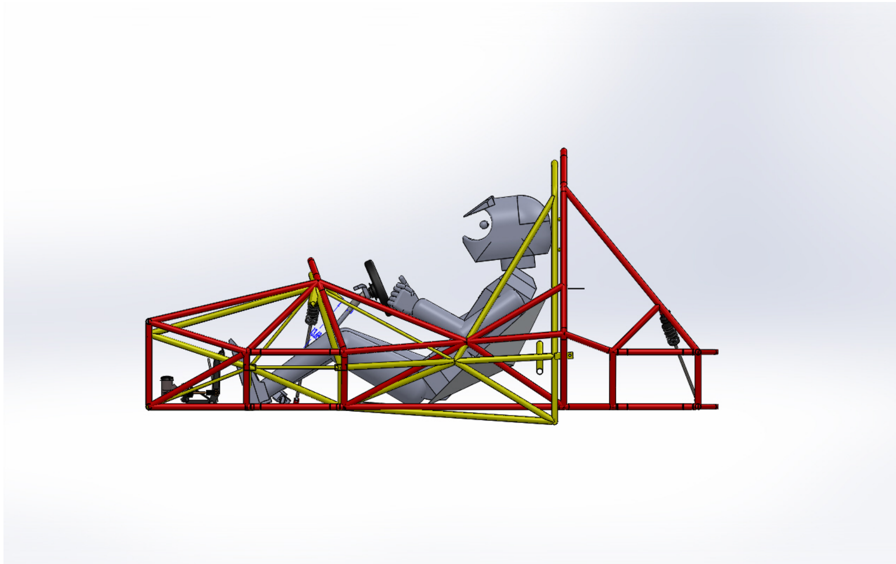


Figure 25 New Frame in Red and Old Frame in Yellow

Final Design 2nd Iteration

After the completion of the first final design iteration several issues were noted. First a rule issues with the side impact members was found.

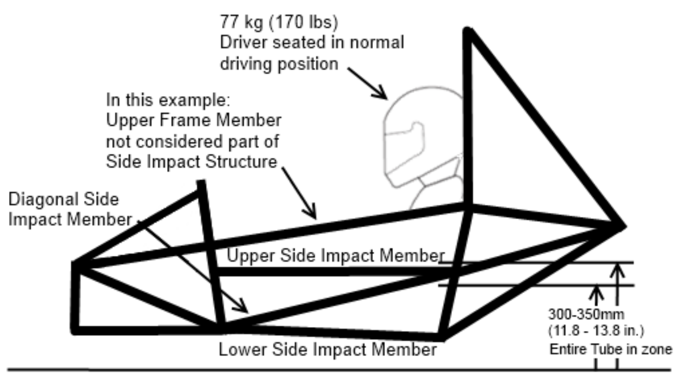


Figure 26 Side Impact Rule

Due to this rule the side impact member was lowered to meet this rule at ride height with a 170 lbs driver. The member was also made parallel with the lower frame rail.

Additional frame rule research was done to make sure the final design met all rules. During the study of example frame documentation on FSAE rules website an issue with the rear hoop braces was found. An explanation of the main roll hoop brace rules is given below.

T3.13.7 *The lower end of the Main Hoop Braces must be supported back to the Main Hoop by a minimum of two Frame Members on each side of the vehicle; an upper member and a lower member in a properly triangulated configuration.*

a. The upper support member must attach to the node where the upper Side Impact Member attaches to the Main Hoop.
b. The lower support member must attach to the node where the lower Side Impact Member attaches to the Main Hoop.

NOTE: *Each of the above members can be multiple or bent tubes provided the requirements of T3.5.5 are met.*

Figure 27 Main Hoop Bracing Rules

Due to this rule the rear braces currently do not meet specifications. Triangulation was added to the rear box sections. This provides the correct support to the main hoop structure. Also as noted during the study of the 2012 frame the issue with the main hoop inclination was resolved by tilting the main hoop rearwards by 9 degrees. This also allowed for more engine space, provided for a better center of gravity, and insured the rules are met under different suspension height configurations. Another performance specification is maintenance. In the final iteration the ability to quickly remove and install the engine was added without the need to pull the engine through the top of the frame. This is due to the possibility of clearance issues with roll brace supports and motor mount supports. The frame rails were widened in the rear to allow the engine to be removed by dropping it out the bottom of the frame.

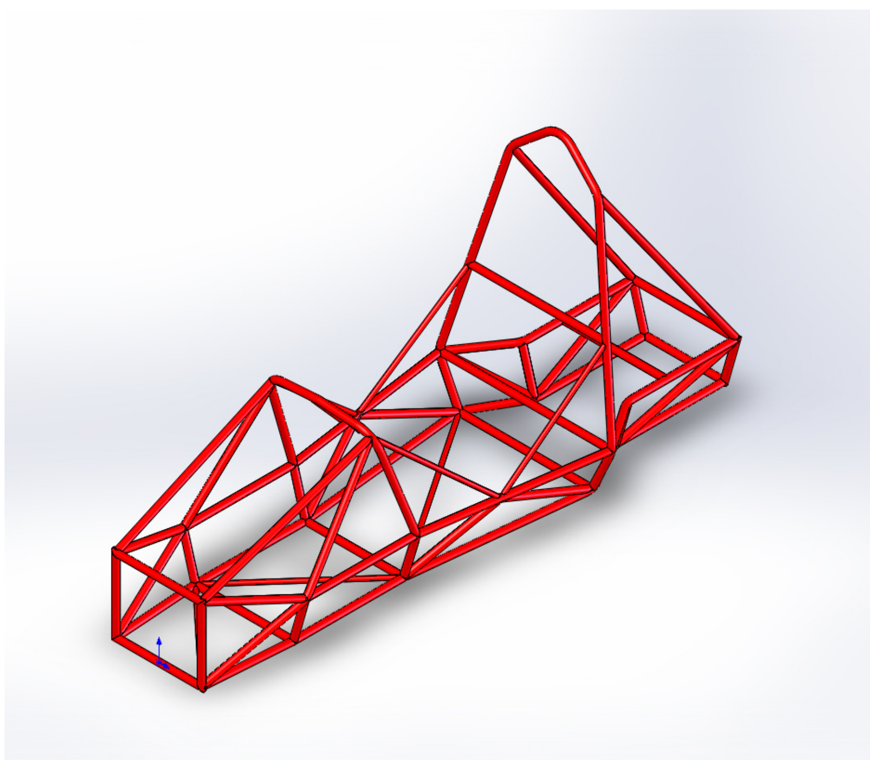


Figure 28 Isometric View of Final Configuration

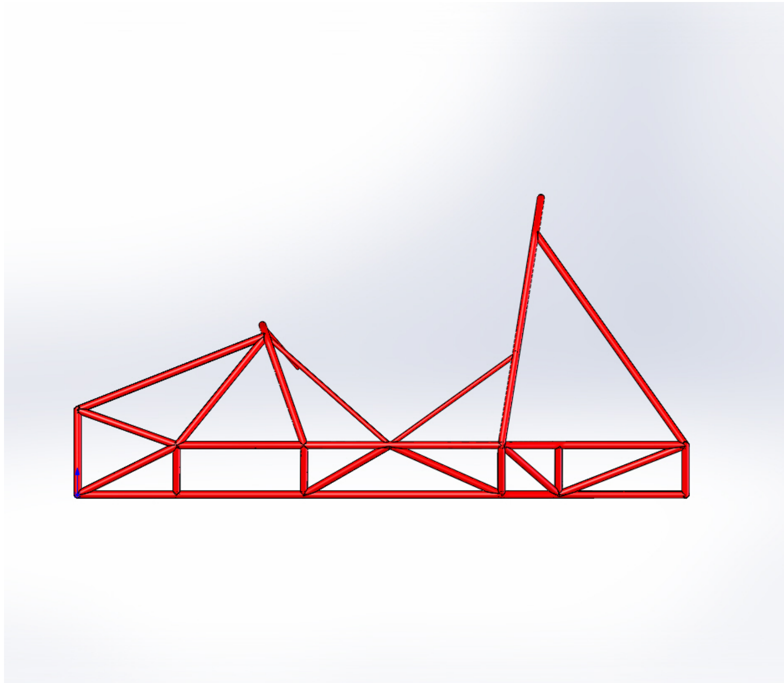


Figure 29 Side View of Final Frame Configuration

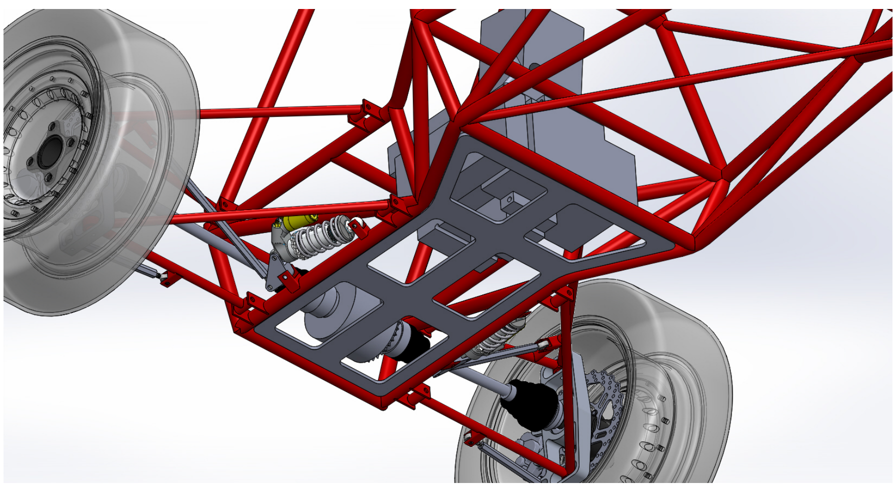


Figure 30 Engine Mount Plate

After the completion of the frame design the suspension geometry was developed. After the suspension geometry is known, the forces on the suspension and frame can be determined. Based on these forces an analysis of the suspension and frame was done.

Specification	Measure
Frame Construction	Tubular space frame
Material	4130 steel round tubing 5/8 to 1 inch dia
Joining method and material	GTAW, ER70-S6 filler
Bare frame weight	64 lbs

Table 7 Frame Specifications

Overview Suspension

The main propose of the suspension is to maintain a constant tire contact patch during different dynamic events. This includes cornering, braking, and accelerating. Last year was the first use of a fully independent suspension on an FSAE car at WPI since around 2008. All previous cars used a live swing axle on the rear suspension. As such a complete back to basics analysis was needed to take place to fully develop and redesign the suspension geometry. This was done to make WPI's cars competitive during dynamic events. Therefore, it was decided to do a complete study of available suspension systems.

To do this, an iterative process was used to aid in designing a completely new suspension system. The following was done

- Concept ranking of different suspension types.
- Complete redesign of geometry packaging.
- Track width and wheelbase selection.
- Packaging components and other design constraints.
- Roll center and camber gain calculation.
- Damper actuation method.
- Selection of dampers.

Study of 2012 Suspension

The previous front and rear suspension on the 2012-2014 car is a conventional double A arm SLA pushrod suspension set up with Cane Creek TTX25 MKII dampers. However, there are several issues with its design. First, the front suspension was originally designed for a swing axle rear end which had allowed the swing axle rear end to roll and handle correctly. Therefore all of the front suspension geometry was not well suited for an independent rear end. One of the main objectives for the design of the front and rear suspension on the new car is to address this issue by redesigning the geometry to match the cars overall setup. Another issue with the front and rear suspension is the camber change setup. The current system uses cam style camber adjustments. However, since each upper suspension arm has a different adjustment the ability to line these up precisely is hard. In addition, changing camber is a time consuming process. As such a better camber adjustment method is important goal for the new car. The rear suspension has several issues. First the suspension was poorly calculated by the 2013 MQP, which caused the car to handle poorly (in addition to the incorrectly setup front suspension). Next, the pushrod setup poorly cleared the drivetrain. This caused the pushrod to need to be bent to allow for clearance for the half shaft axles. This weakened them and caused the rod end to be slightly placed in bending. The goal will be to avoid this issue by redesigning the damper actuation method.

Objective

The main goals for the redesign of the 2015 front and rear suspension setup are to address the known issues found in the 2012-2014 car.

- Must be simply adjustable using standard automotive tools.
- Address camber adjustment issues.
- Calculate suspension dynamics for tuning.
- Critical mounts must be in double shear to minimize shear stresses
- No rod end can be placed in significant bending.
- Must clear drivetrain.
- Redesign damper actuation method
- Low roll center (under 3 inches and non-negative) and reasonable camber gain (0.5 deg per 1 deg roll).

Research

The first objective was to do detailed research into different suspension types. 6 different types of suspensions were studied. These included MacPherson, live axle, trailing arm, and double wishbone.

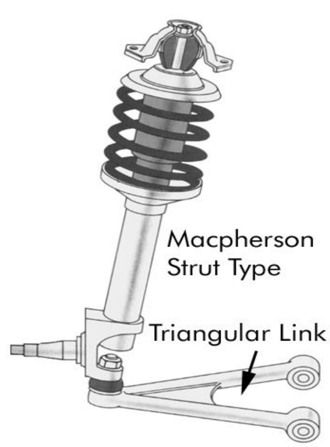


Figure 31 MacPherson

The first suspension type studied was MacPherson strut. This type uses a single link directly connected to the shock. This provides a very simple and cost effective suspension setup. This makes it one of the most popular suspension setups on modern cars. There are several issues. First, this suspension type provides no camber gain control. Second it has poor adjustability and only has a single load path into the frame.

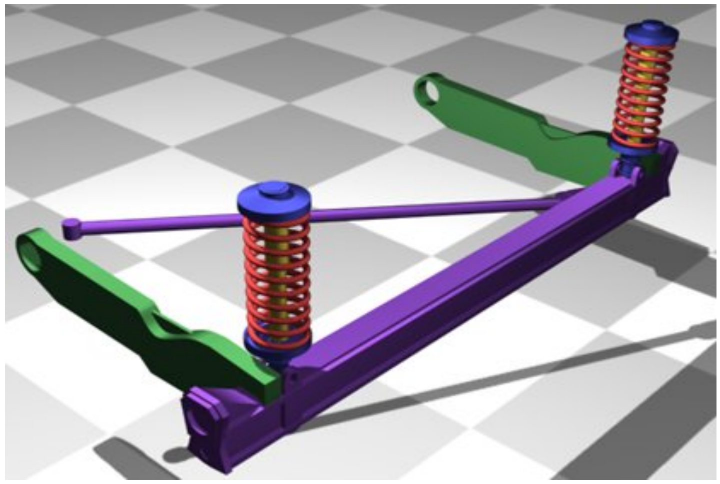


Figure 32 Live Swing Axle

Next was a swing axle. This type of axle can only be used on the rear of the car. It provides a very simple and cheap suspension system. It can provide great off the line traction since it does not have squat during load transfer. However this axle has no roll ability. Therefore it has no camber control and poor tire patch control. This makes it unpredictable during cornering and does not provide the same performance as independent setups.

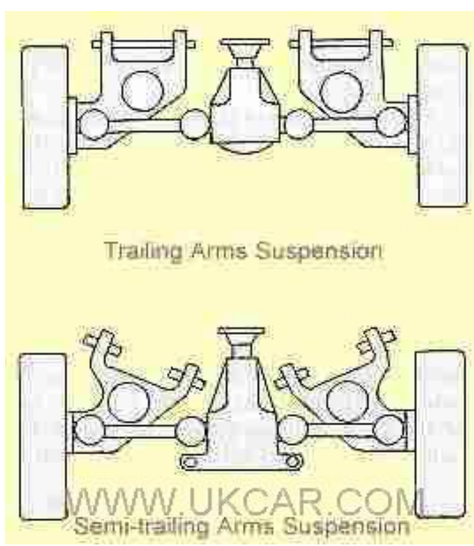


Figure 33 Trailing Arm

A trailing arm setup provides a simple independent rear suspension with few parts and great strength. It allows for roll and independent movement of each wheel without affecting the opposite side. However, it places all forces on a single pivot point. It is also hard to build in adjustability into a trailing arm setup since the arms are typically solid pieces. This suspension also has no camber gain control.

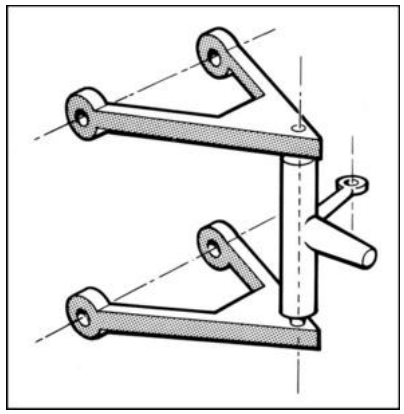


Figure 34 Double Wishbone Suspension

Double wishbone provides an excellent way of controlling wheel travel. They provide a simple and highly adjustable setup. The system can be setup in a number of different ways including equal length and unequal length. This allows for a great range of camber gain options. The system does use more parts than a convention MacPherson setup or a trailing arm suspension, but the adjustability of the setup is a great design trade off. This is by far the most used set up on FSAE cars.

By using right triangle shaped control arms on the rear of the car, it is possible to offset the centerline of the drive axles from the suspension A-arms. This allows for more clearance to the pushrod to attach to the lower suspension arm. As seen below several teams have uses this setup. It also allows for the rear suspension to be shorted therefore saving additional weight. This is done by mounting the differential off the end of frame.

Figure 35 Example of Right Angle Control Arms

Design

Preliminary Design Concepts

It was decided after research was completed that any live axle or swing axle suspension would not be considered since it would not be in line with the design goal of the whole car. Therefore, four different suspensions were chosen to move forwards with. These are double wishbone, unequal length double wishbone, trailing arm, and MacPherson Strut.

Concept 1: Double Wishbone

This is the simplest form of double wishbone suspension. It uses equal length arms and has zero camber gain during suspension travel. This provides a simple method of mounting arms to frame since it is easier to package into the car. However, the performance is not the same as an unequal length setup. This is due to its inability to gain camber during suspension travel (roll). This does not allow it to keep a constant tire contact patch with the ground.

Concept 2: Unequal Double Wishbone

This is an extension of the previous suspension. This uses a shorter upper control arm to allow camber gain during roll. By adjusting these arm lengths the gain vs wheel travel can be adjusted to whatever is necessary for the roll rates and roll centers of the car.

Concept 3: Trailing Arm

This system provides a fixed camber gain during wheel travel (or none depending on setup). It also has a high un-sprung weight compared to other systems. However, it is very simple and strong. It does place large amounts of stress onto frame mounting points compared to other systems as well.

Concept 4: MacPherson Strut

While easily one of the simplest setups but also difficult to manufacture due to custom parts needed to mount dampers and lower suspension arms. It also provides subpar performance compared to unequal length suspension setups.

Concept Selection

After research was completed, a design matrix was completed to rank each setup. These were then weighed based on decision factors.

Decision factor descriptions	
Ease of Access	How easy is to reach adjustments and to remove components
Fatigue Resistance	How much stress are components under during operation
Strength of Design	How much information is available to provide for a strong analysis of design
Simplicity	How few moving parts does the system use or how complex is a single item
Ease of Machining	How easy will it be to manufacture, are off the shelf parts available
Cost	How expensive will this design be compared to others
Compatibility	How much room does this design take up and how well will it integrate with other subsystems
Serviceability	How easy is it to replace parts and retain adjustments

Manufacturability	How many manufacturing methods will be required to develop the final suspension
Performance	How well will this suspension meet the final performance goals
Tire Patch Control	How well does this type of suspension maintain camber, toe, caster under dynamic events
Adjustability	How much adjustability is built into the suspension compared to others

Table 8 Design Factor Decisions

		Concept 1		**Concept 2**		**Concept 3**		**Concept 4**	
Decision Factor	**Weight**	**Semi-Trailing Arm (Rear Only)**		**Double Wishbone**		**Unequal length wishbone (SLA)**		**MacPherson Strut**	
		Score	Value	Score	Value	Score	Value	Score	Value
Ease of Access	6	9	54	9	54	9	54	8	48
Fatigue Resistance	8	7	56	8	64	8	64	6	48
Strength of Design	7	8	56	8	56	8	56	7	49
Simplicity	9	9	81	6	54	6	54	8	72
Ease of Machining	6	7	42	6	36	6	36	8	48
Cost	7	8	56	5	35	5	35	8	56
Compatibility	8	8	64	7	56	7	56	8	64
Serviceability	6	9	54	7	42	7	42	7	42
Manufacturability	7	7	49	7	49	7	49	8	56
Performance	10	6	60	8	80	8	80	6	60

Tire Patch Control	8	5	40	7	56	9	72	5	40
Adjustability	7	1	7	7	49	7	49	6	42
Totals			**619**		**631**		**647**		**625**

Table 9 Design Matrix

As seen above the clear winner was the unequal length double wishbone suspension. This provides the best mixture of performance and simplicity. It also is the setup currently being used on the car so the team already has some experience working with it.

Final Design 1st Iteration

After a detailed review of all of the above information, a mockup of the suspension was drawn in SolidWorks using 3D sketches. The front suspension closely resembles the current setup with different arm lengths and a slightly wider track. The next difference is the offset rear control arms, which place the rear drive axles behind the rear frame. This allows them to have a clear run to the differential. It also allows the pushrods to directly mount to the lower control arm without any clearance issues.

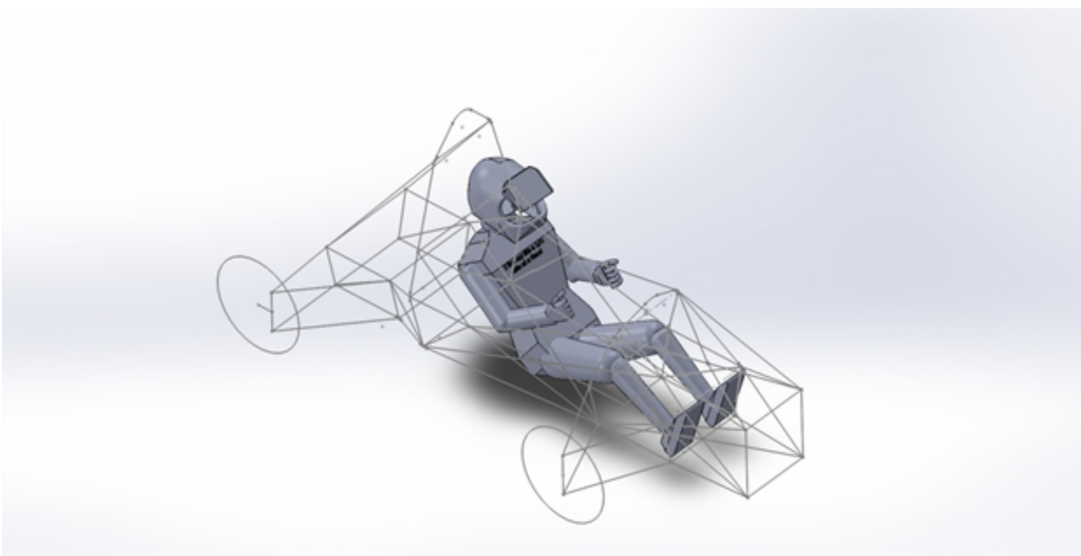


Figure 36 3D Mockup of Suspension

By manipulating 3D sketches in SolidWorks, two methods of shock actuation were explored, pushrod actuation and direct acting. It was determined that pullrod actuation was not feasible due to the high offset rear arms and packaging of the shocks below or above the driver's legs. The chassis design does not permit this option since there would not be enough space when passing the FSAE templates through the chassis. The pushrod design results in a very long pushrod actuating a rocker mounted on the top forward facing chassis members. As such, a direct acting setup was chosen for the front suspension. The same concept was applied to the rear suspension as well since it makes for a much more simple method of actuating the dampers. Seen below is the current direct acting damper setup. This design decreases mechanical complexity and weight.

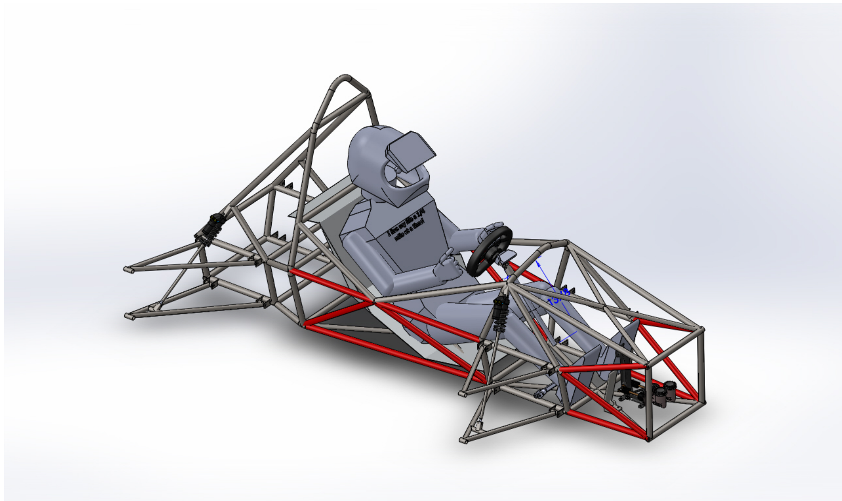


Figure 37 Overview of Suspension with Direct Acting Shocks

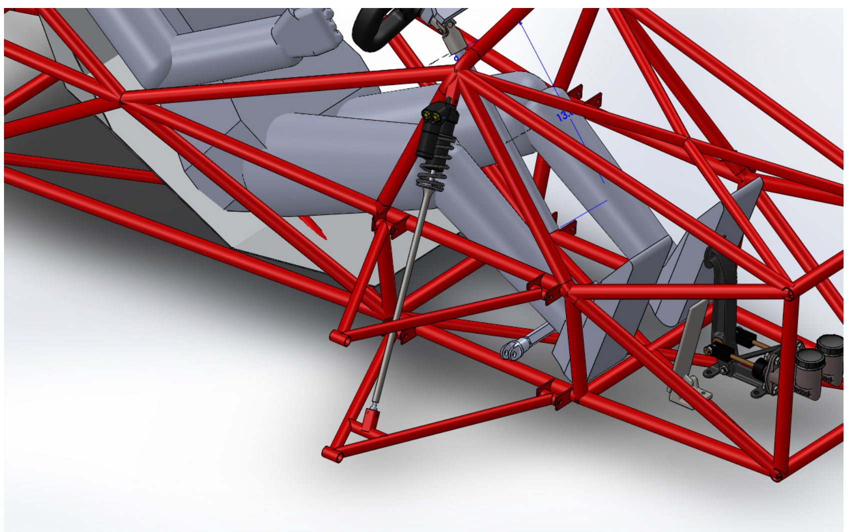


Figure 38 Detailed View of Front Suspension

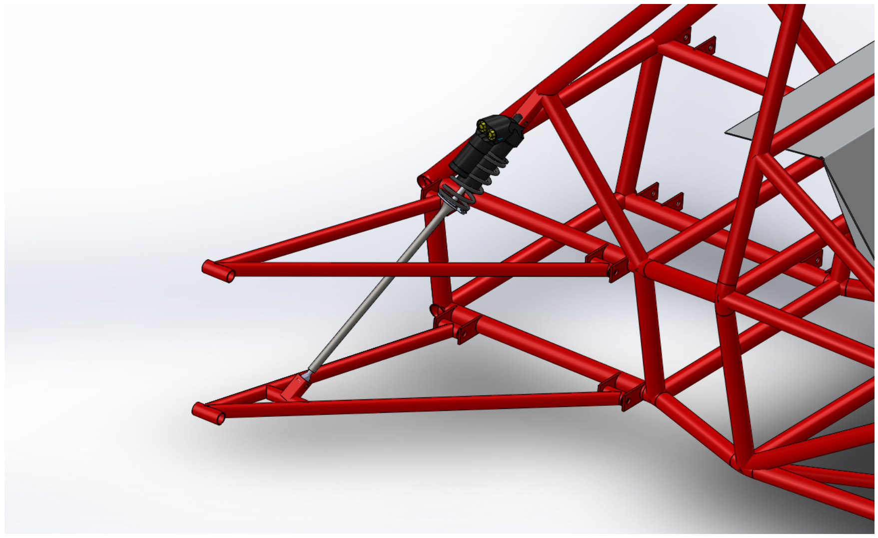


Figure 39 Rear Suspension Detailed View

Final Design 2nd Iteration

After the first iteration was completed several issues were noted. First loading issues were foreseen with the offset uprights required for drive axle clearance. As such it was decided to lengthen the frame slightly and move the differential back inboard. This also helped strengthen the rear control arms while improving packaging of the rear driveline. The new control arm geometry also opened up more options for damper methods. This also made it possible to package a pullrod or pushrod setup into the rear of the car.

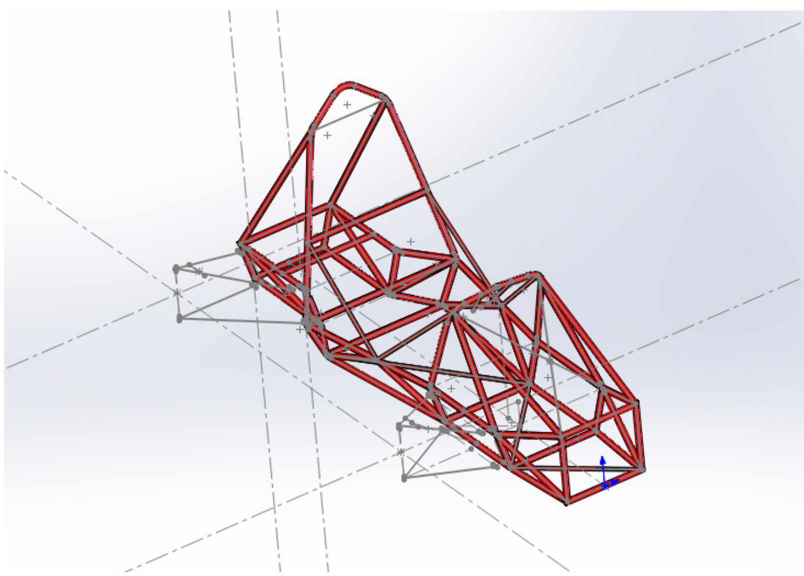


Figure 40 Suspension Arms Final Configuration

The control arms are constructed from 5/8 inch tubing with a 1/16 inch wall thickness chromoly tubing. To allow for steering of front uprights and adjustment of toe in rear the ends feature spherical ball bearings. These are installed in Uniball cups which use snap ring retainers and are welded onto the arms. The inner joints use standard bronze bushings with are press fit into the chromoly tubing.

Figure 41 Uniball Spherical

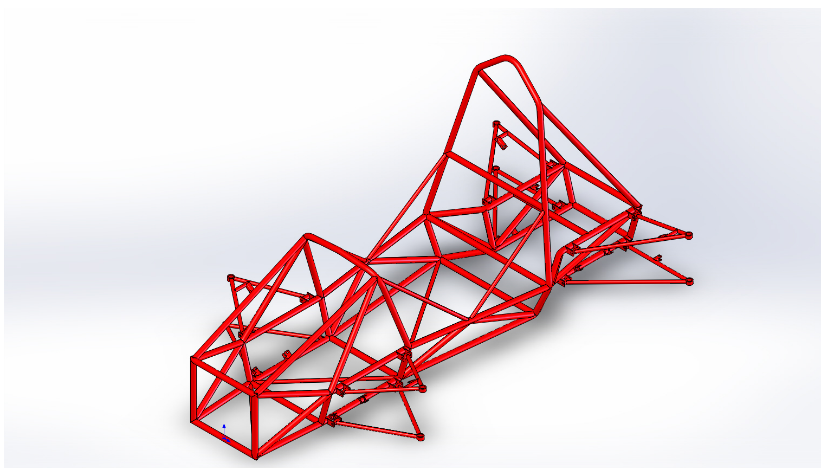


Figure 42 Control Arms

The next goal was to improve the damper actuator method. With the new control arm design the actuator method was switched back to a rod setup. The choice was made to use a pullrod setup. This method places the rod in tension instead of compression while also lowering the center of gravity of the car due to the low mounted dampers. Several mounting methods were explored until a horizontal laying damper was selected. This provides the best overall packaging. The rocker is of very simple design to allow for easy manufacturing. The rod itself is constructed from an off the shelf turnbuckle and spherical rod ends. Mounting to the frame and control arms is done via square tubing sections.

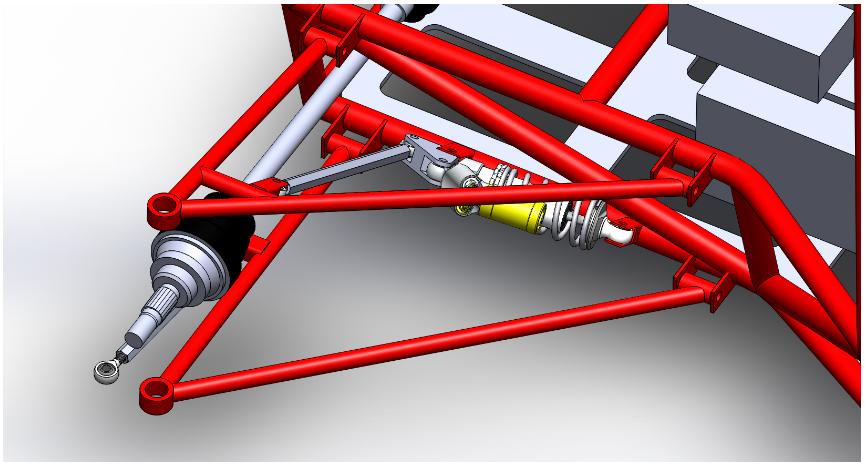


Figure 43 Rear Pullrod

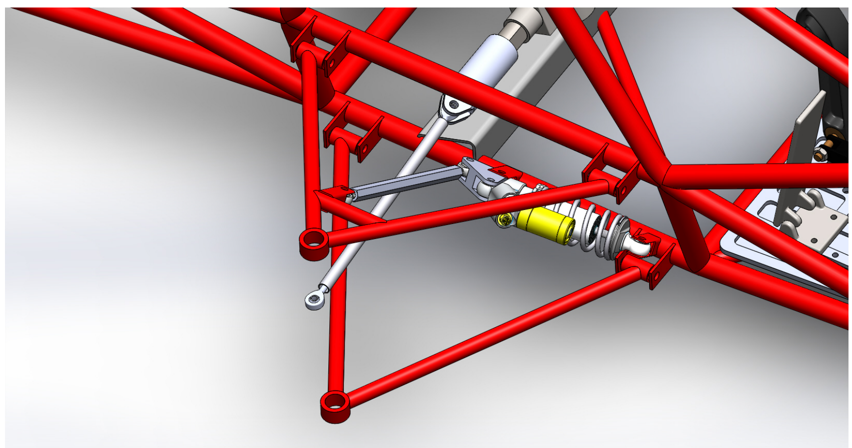


Figure 44 Front Pullrod

Not pictured here is camber adjustment. It was decided to use a shim style camber adjustment method. This is done by placing shims behind the upper control arms mounting tabs. This spaces the upper control arm in or out depending on the desired static camber. Static camber is set to 1.5 deg front and 0.5 deg rear.

To complete the analysis of the suspension the roll centers were determined. Based on this data and suspension geometry the loading was determined for different dynamic events. These values were then used for the final FEA analysis.

Specification	Measure
Track Width	50 inches
Wheelbase	61 inches
Travel	1 inch bump and 1 inch droop

Camber Static	1.5 degrees front and 0.5 degrees rear
Anti dive and Squat	0
Dampers	TTX25 MkII - Ohlins

Table 10 Suspension Specifications

Suspension and Pullrod Analysis

Estimated Weight Distribution and Center of Gravity

For the new frame and suspension to be designed and tested before manufacturing, a full understanding of the forces and loads seen during different dynamic events needs to be known. By knowing this information, designs can be analyzed and improved. An FSAE car is a highly dynamic system. Forces seen in different components of the car under acceleration, braking, and cornering vary greatly during its operation. To test the new suspension design and frame during these dynamic situations several finite element analysis (FEA) simulation models were used. To make these models as accurate as possible, the loads created during these dynamic events must be known.

To find the suspension loads, the first task was to understand the car's weight distribution. The weight distribution is very important since it allows for the center of gravity (CoG) to be found. Since the new car does not exist yet the current car was used. Using corner weight scales, the car was weighed with a driver to find the corner weights on each tire. These measurements were taken from the 2014 FSAE MQP report.

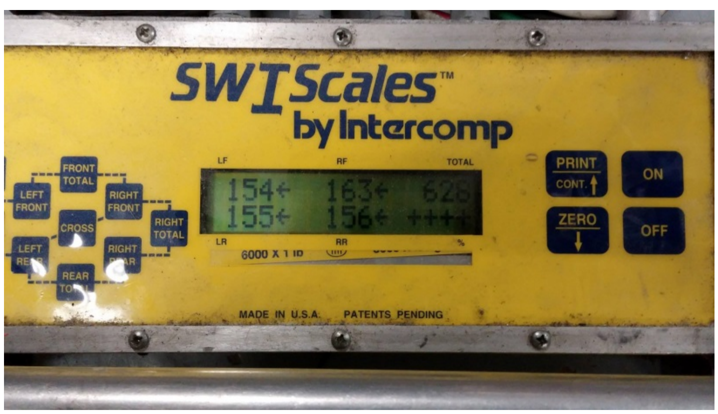


Figure 45: 2014 FSAE Car Corner Weights

Table 11: Corner Weights

Corner Weights	Left	Right
Front	154lbs	163lbs
Back	155lbs	156lbs
Total		628lbs

As seen in the table above, the corner weights are very close to 50% weight distribution front to rear. This is optimal since it allows for more natural handling and better responsiveness. For the analysis, the weight will be rounded to a perfect 50/50 weight distribution since driver weight can vary. This allowed the CoG to be found on the car's centerline. The overall CoG is centered between the front and rear wheels. Since the wheelbase of the new car is 61 inches, this placed the CoG at 30.5 inches behind the front wheels. These measurements were confirmed with Suspension Calculator by Cristobal Lowery. This software package is able to perform a wide variety of load-based suspension calculations including 3D loads on double wishbone suspensions. In the figure below, the longitudinal CoG was calculated. It was found to be 0.775m from the front wheels which is exactly 30.5118 inches.

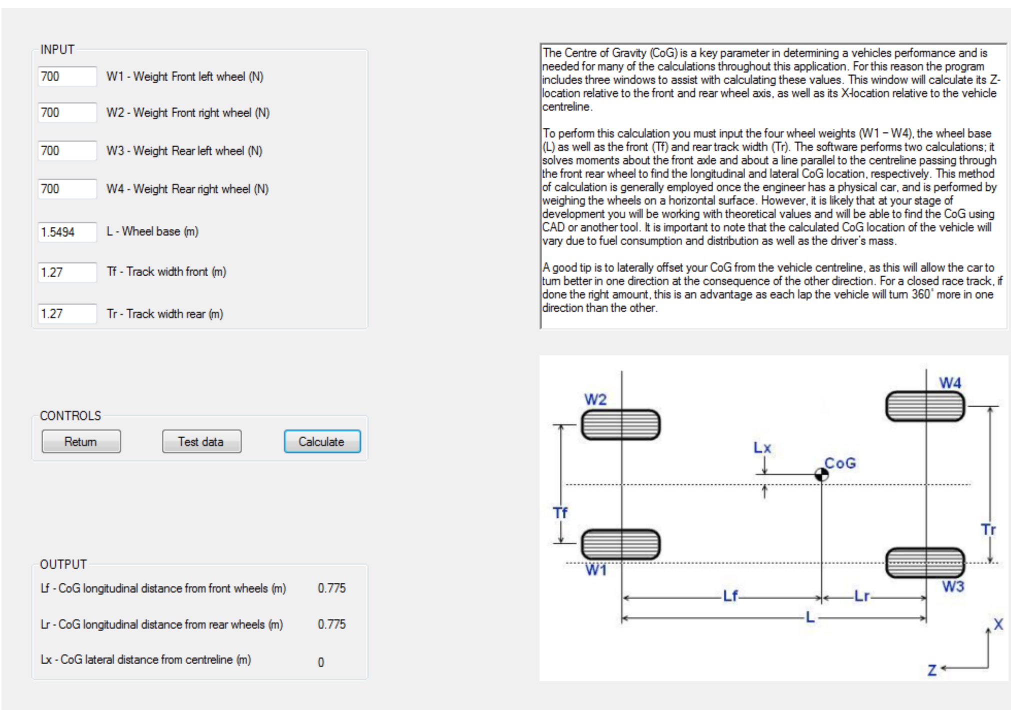


Figure 46: Static CoG Location

However, the height of the CoG was still not known. To find the height of CoG, the car needed to be raised at the rear wheels while the load of the front wheels was measured. Then, by using the angle between the car and the ground, the height of the CoG can be determined.

To do this, the corner weight scales were placed under the front wheels and the car was jacked up to 5.27° from horizontal. Once again, by using Suspension Calculator, the CoG height was determined to be 0.252m or 9.92126” from ground.

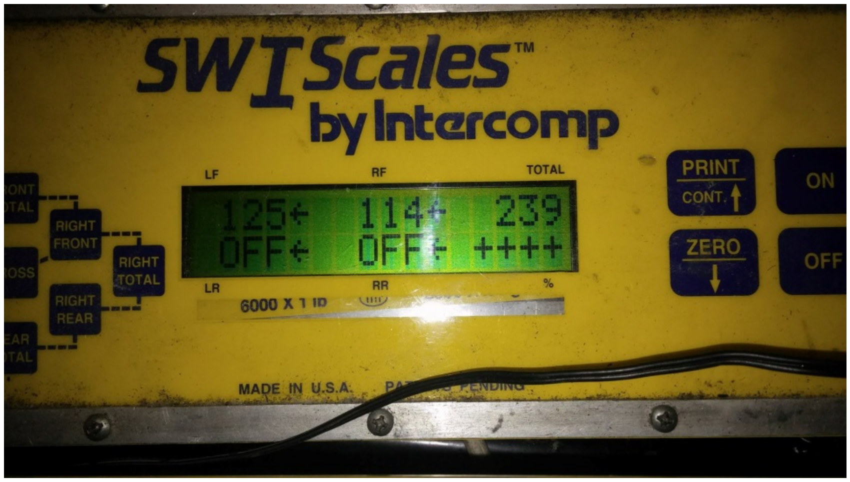


Figure 47: Front Weight at 5.27°

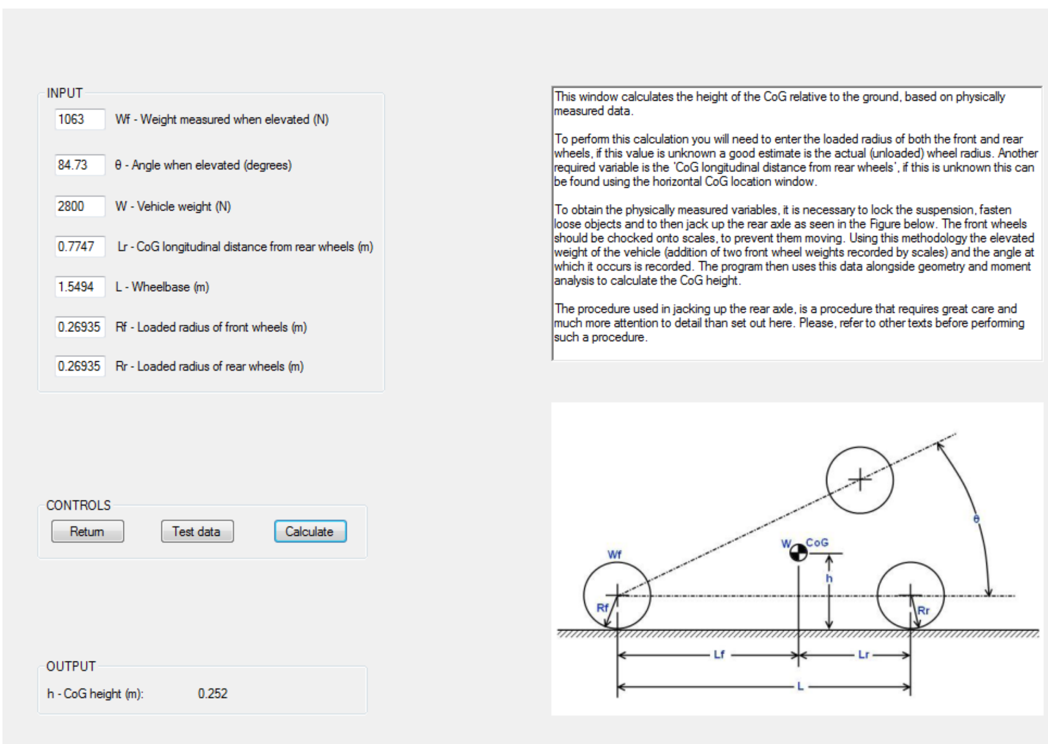


Figure 48: Height of CoG

Roll Center

With both the longitudinal position and height of the CoG known, the next step was to determine the roll center of the suspension geometry. The roll center of a vehicle is the notional point at which the cornering forces of the suspension are reacted to the frame. The roll center is very important to the loading of the suspension while cornering. For this, Vsusp Suspension Geometry Calculator was used. This tool allows for the suspension geometry data to be input and the software creates a virtual model of the suspension. Using measurements from SolidWorks, the front and rear suspension were replicated into this tool. The final roll center was found to be at 65mm above the ground or 2.6 inches at estimated ride height for the front and 13mm or .512 inches for the rear. While the ride height of the vehicle may be lowered to decrease this roll center, the given calculation will provide a good worst case scenario for suspension loads.

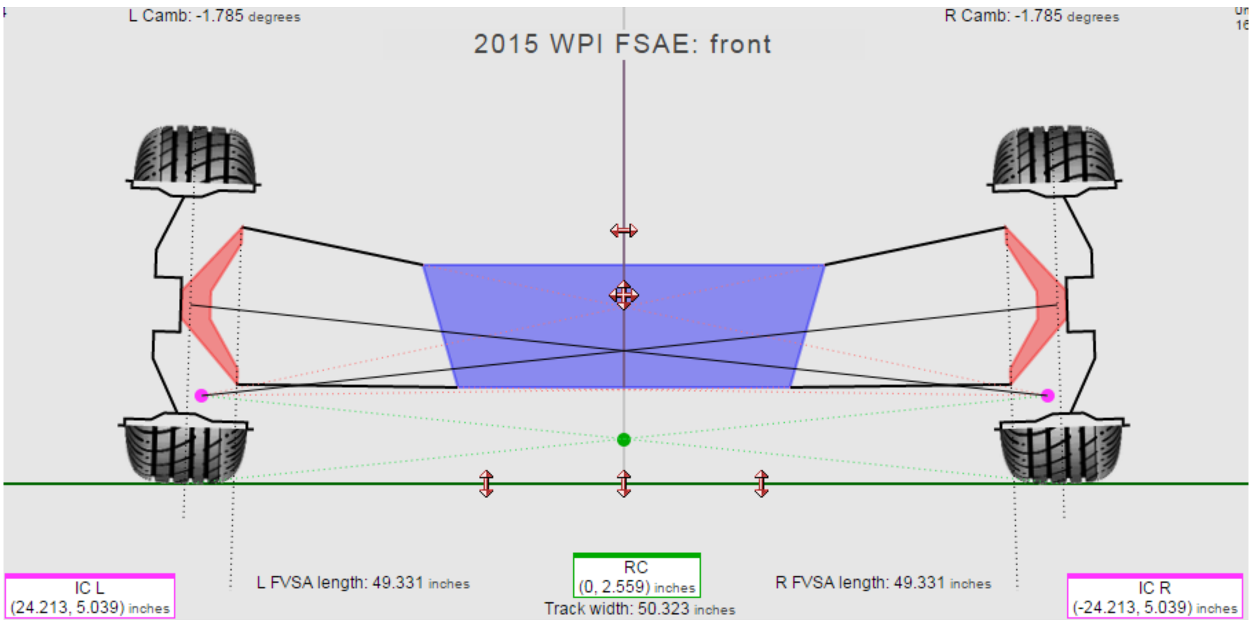


Figure 49: Front Roll Center

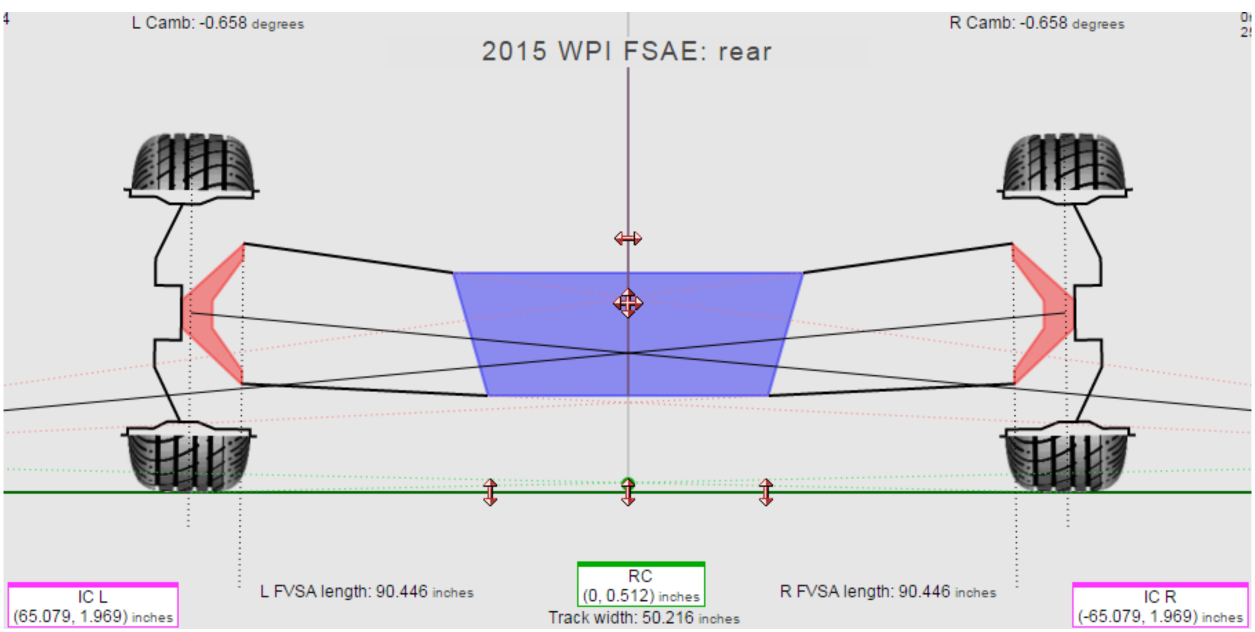


Figure 50 Rear Roll Center

Travel (Inch)	RC Front	RC Rear
+1 (bump)	2.09	0.08
0	2.56	0.51
-1 (droop)	3.11	0.945

Table 12 Roll Centers

Camber Gain

Another key factor to the suspensions performance is the camber gain rate. Camber gain is the difference of the camber angle after suspension travel, typically 1 inch of suspension bump and 1 inch of droop or by chassis roll. Below the camber gain curves are given for both the front and rear suspensions.

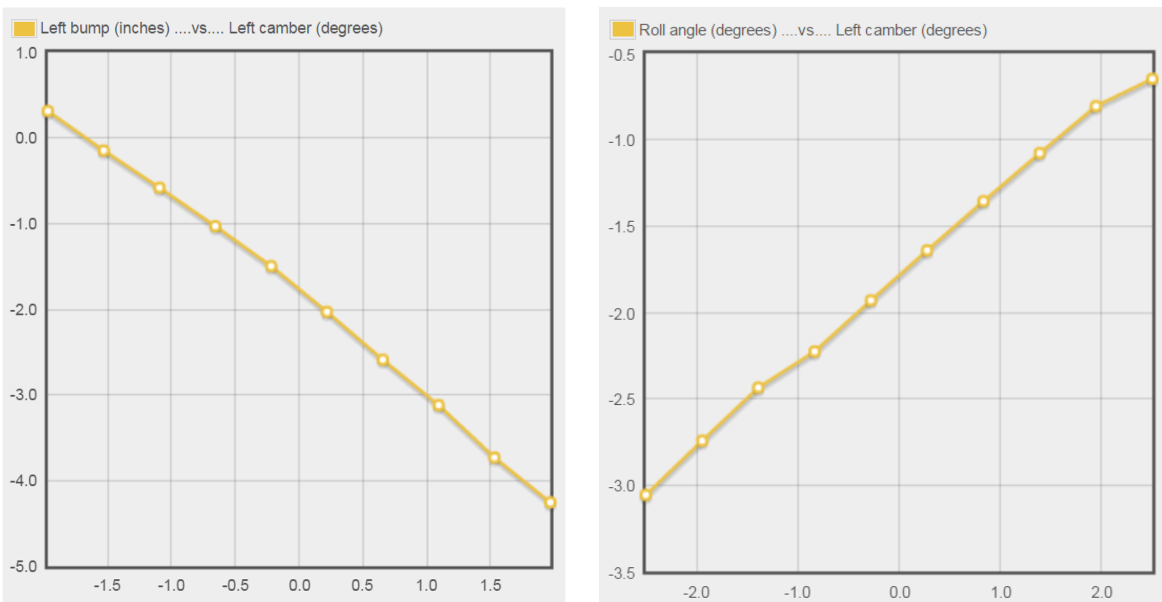


Figure 51 Front Suspension Camber Gain (X-axis Travel/Roll Angle, Y-axis Camber Angle)

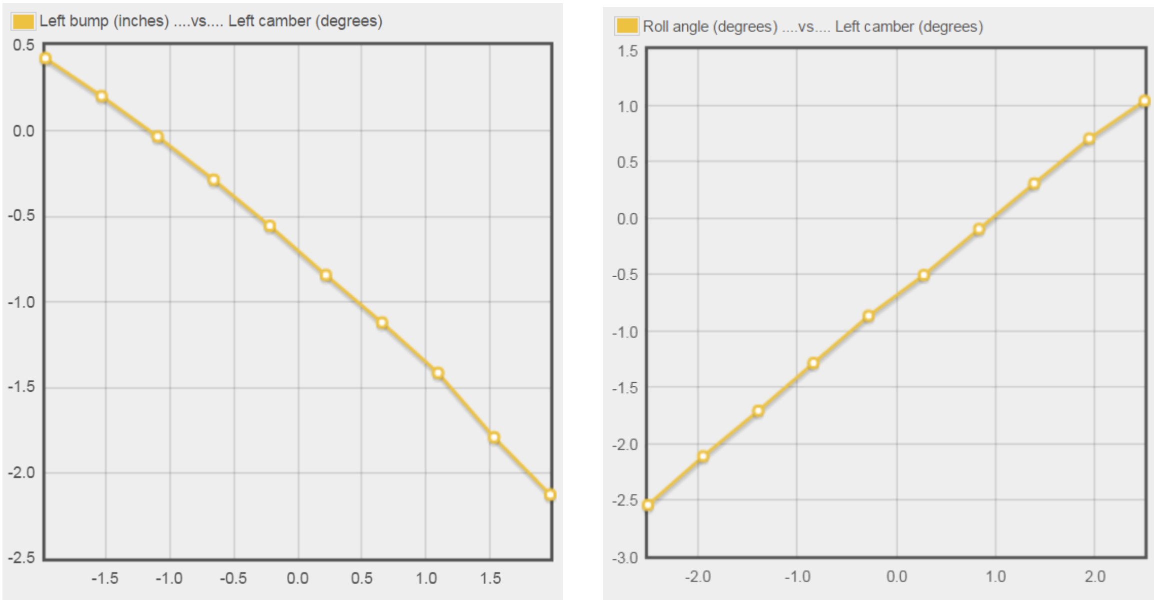


Figure 52 Rear Suspension Camber Gain (X-axis Travel/Roll Angle, Y-axis Camber Angle)

Travel (Inch)	Camber Front	Camber Rear
+1 (bump)	-2.958	-1.385
0	-1.785	-0.658
-1 (droop)	-0.669	-0.101

Table 13 Camber Gain

Analysis of Final Design: Suspension Dynamics and Loads

Dynamic Event Forces

To fully test the design of the new rear suspension, the forces applied to the car during different dynamic events needed to be calculated. For the purpose of these simulations, four different dynamic event situations were calculated. These were during 1.5g braking, 1.5g lateral load, and 3.5g of suspension bump. These values were taken from *Race Car Vehicle Dynamics*, as recommended dynamic testing situations. By calculating these three different events, the highest possible loads on each component can be found and can be used for FEA simulations. All dynamic suspension calculations were completed using Suspension Calculator.

Dynamic 1.5g Braking

The first step to determining the loads placed on the rear suspension during 1.5g of braking is to find the load transfer caused by the deceleration of the car. This is where the CoG values were used.

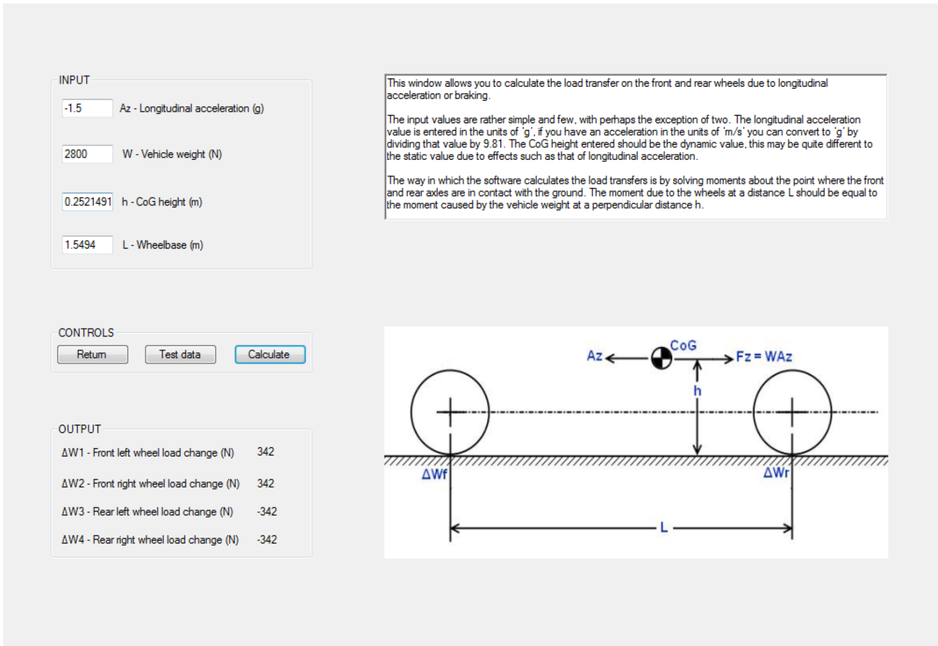


Figure 53: Load Transfer during 1.5g Braking

During a hard braking situation, the front wheels will see an increased load of 342N while the rear wheels will be unloaded by 342N. To find the total wheel load, the static load must be added to this load transfer for the front wheels and subtracted for the rear. Under a 1.5g braking event, the final rear wheel load is 1042N front and 358N rear.

Equation 1

$$Wheel\ Load\ =\ Load\ Transfer\ +(Static\ Weight/4)$$

$$1042N\ =\ 342N\ +\ (2800N/4)$$

$$358N\ =\ -342N\ +\ (2800N/4)$$

After the wheel load had been determined, the next step was to calculate the suspension loads. Suspension Calculator has a built in tool that calculates the forces on each suspension joint in X, Y, and Z coordinates. To use this tool, first the suspension joints' coordinates needed to be determined. These values were obtained from the SolidWorks model.

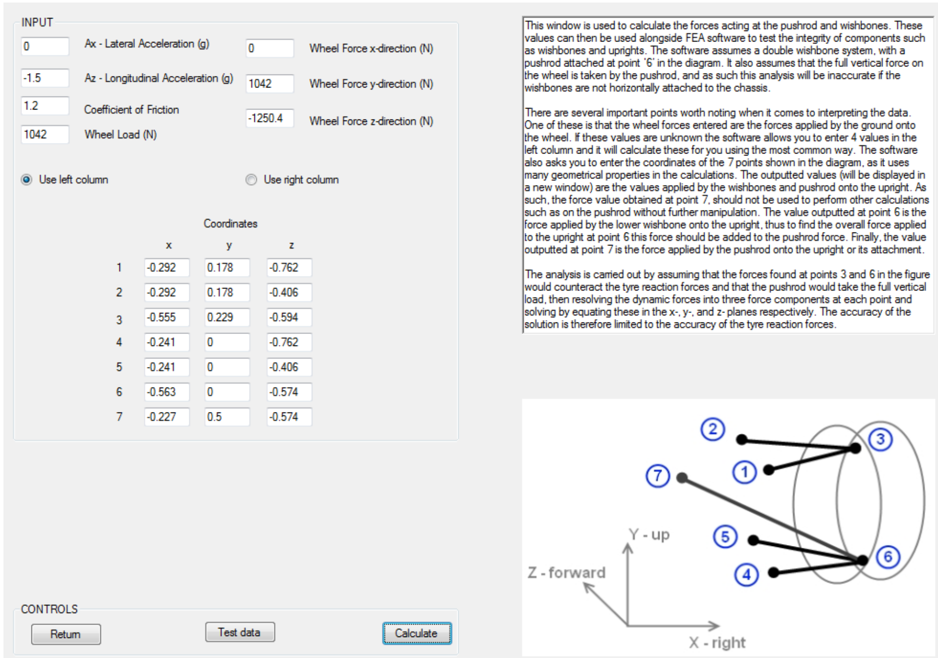


Figure 54: Front Suspension Coordinates

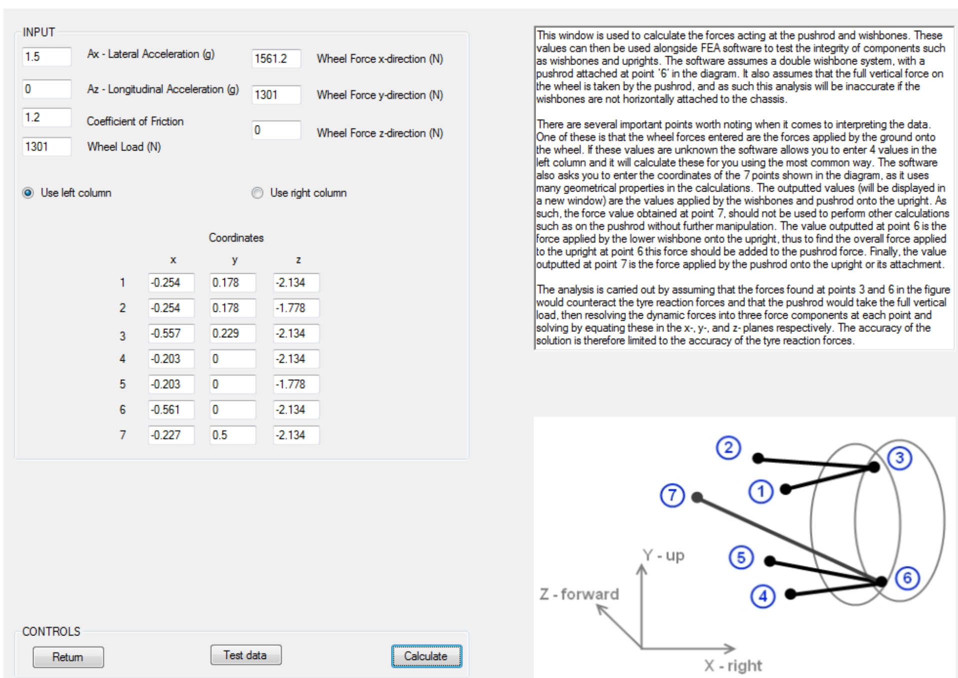


Figure 55 Rear Suspension Coordinates

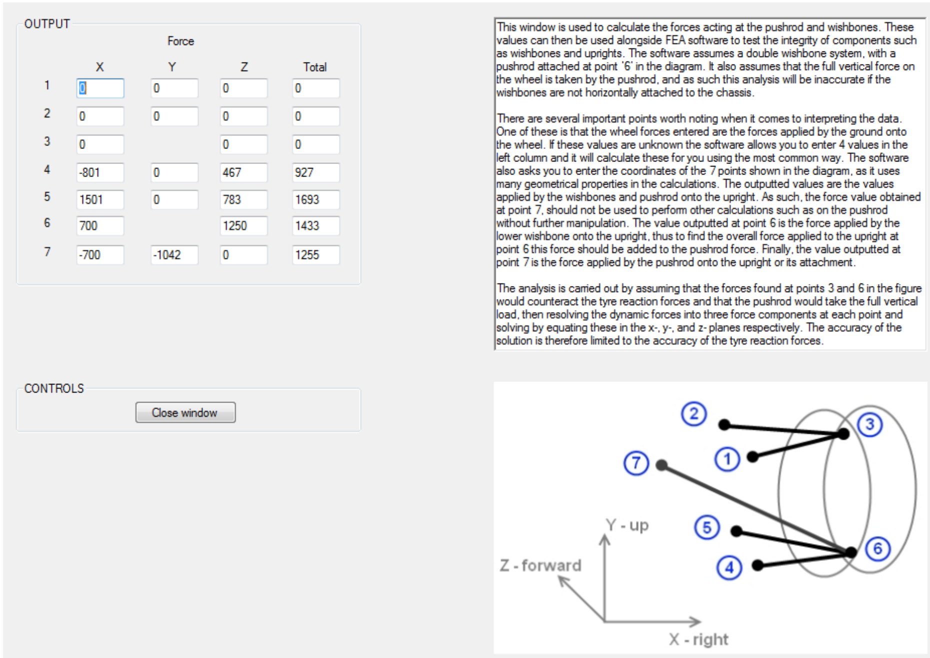


Figure 56: Front Suspension Arm and Pushrod Forces Under 1.5g Deceleration

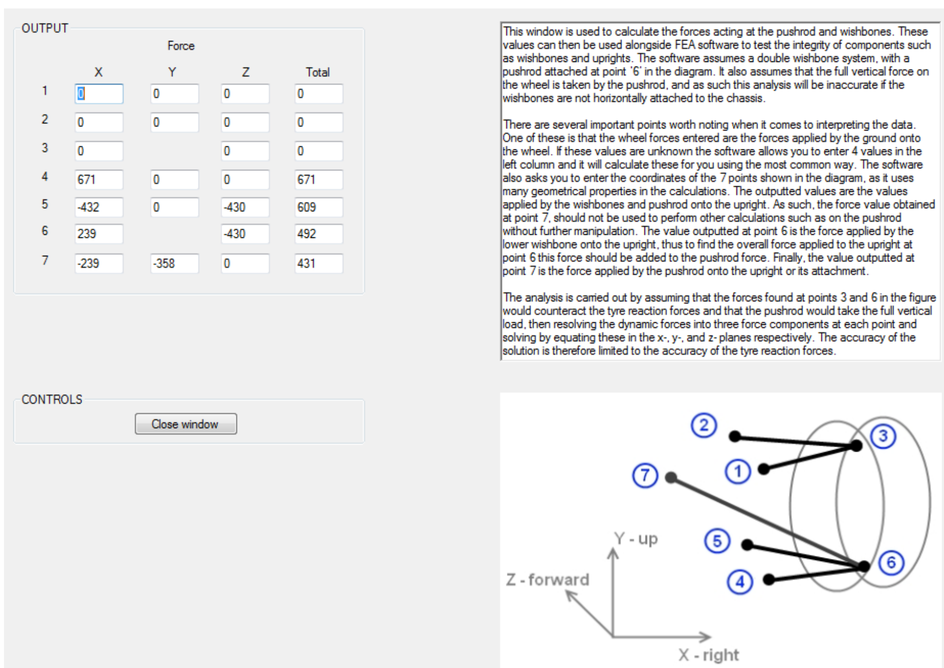


Figure 57 Rear Suspension Arm and Pushrod Forces Under 1.5g Deceleration

The 1.5g braking wheel suspension forces are seen in figures above.

Dynamic 1.5g Lateral Load

The first step to determining the loads placed on the rear suspension during 1.5g of lateral load is to find the load transfer caused by the lateral acceleration of the car. Once again, the previously calculated CoG values were used.

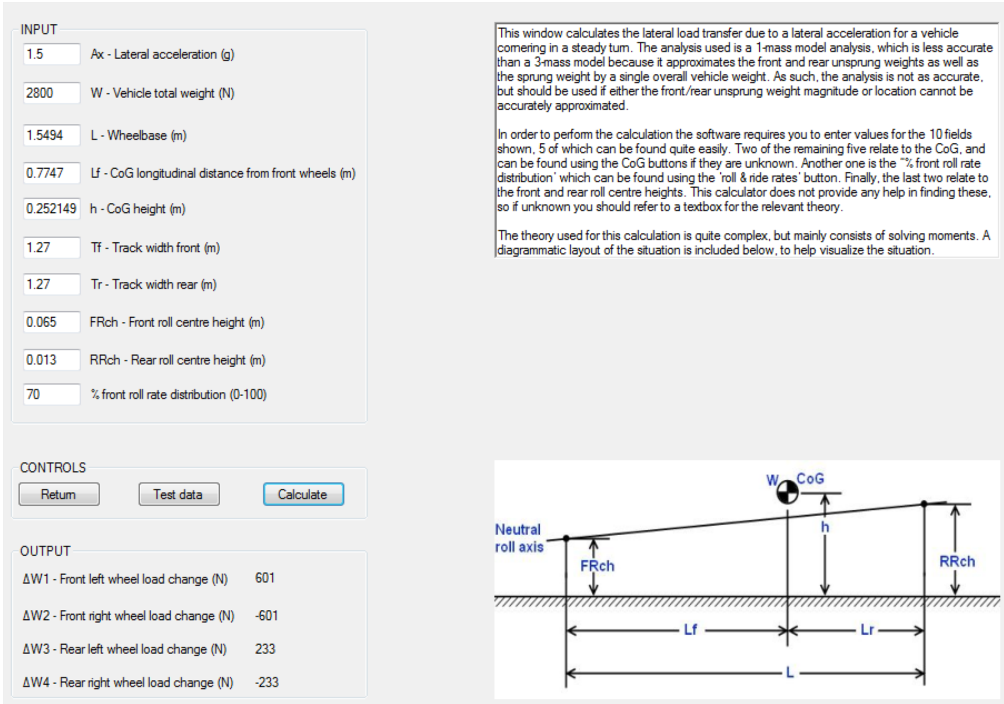


Figure 58: Load Transfer during 1.5g Lateral Load

The calculator requires several more inputs for this calculation. All variables have been determined except for front roll rate distribution. This is calculated by using the front and rear spring rates along with total suspension travel front and rear. During a hard cornering event, the inside wheels experience decreased loading while the outside wheel loads are increased. The rear wheels see a load transfer of 233N from right to left while the front wheels see 601N. To find the total wheel load, the static load must be added to this load transfer. Under a 1.5g lateral event, the final rear wheel load is 933N and front is 1301N.

$$Wheel\ Load\ =\ Load\ Transfer\ +(Static\ Weight/4)$$

$$933N\ =\ 233N\ +\ (2800N/4)$$

$$1301N\ =\ 601N\ +\ (2800N/4)$$

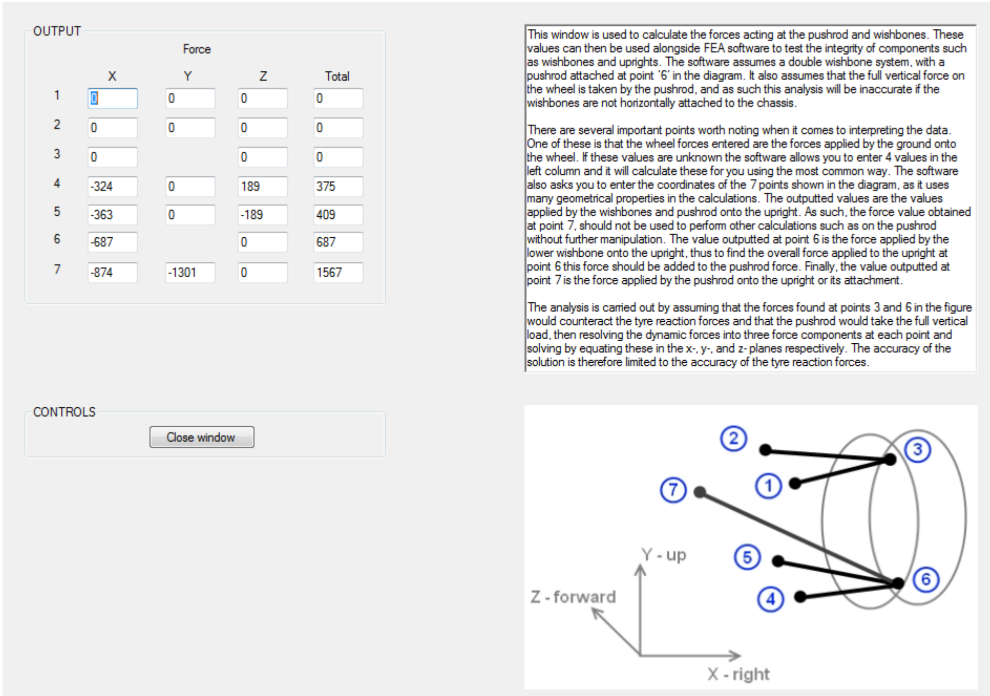


Figure 59: Front Suspension Arm and Pushrod Forces under 1.5g Lateral Acceleration

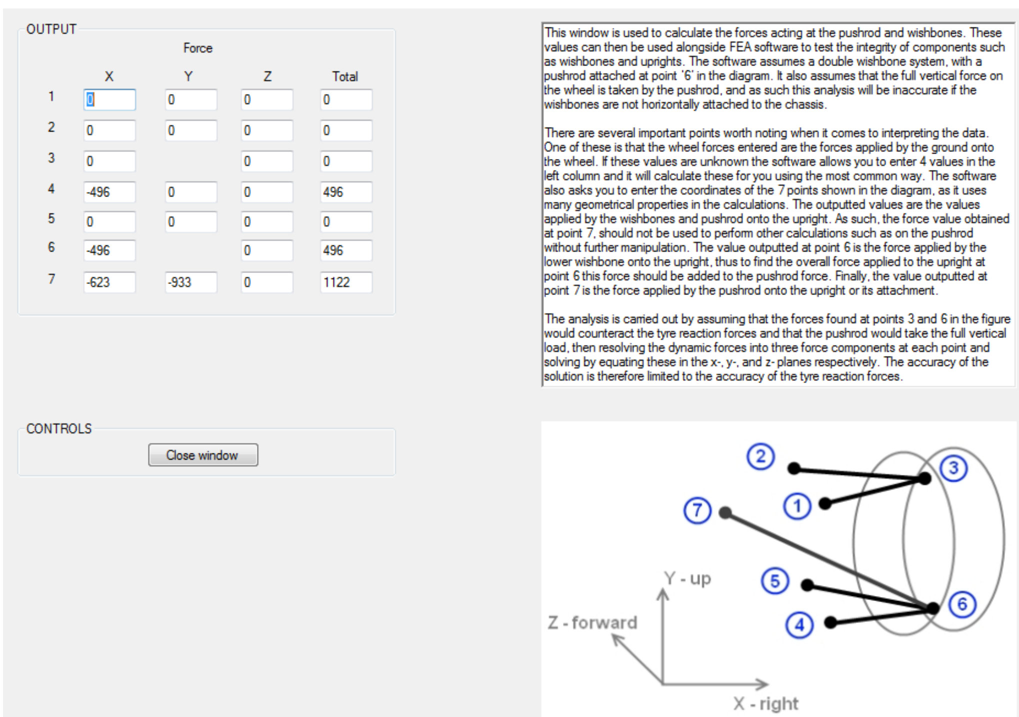


Figure 60 Rear Suspension Arm and Pushrod Forces under 1.5g Lateral Acceleration

The 1.5g lateral acceleration forces on the suspension are seen in figures above. These loads are larger than the 1.5g acceleration forces.

Dynamic 3.5g Bump

The first step to determining the loads placed on the rear suspension during 3.5g of suspension bump is to find the wheel load caused by this upward wheel acceleration. The only needed value for this calculation is the unsprung wheel weights. These were found by weighing suspension components of the car.

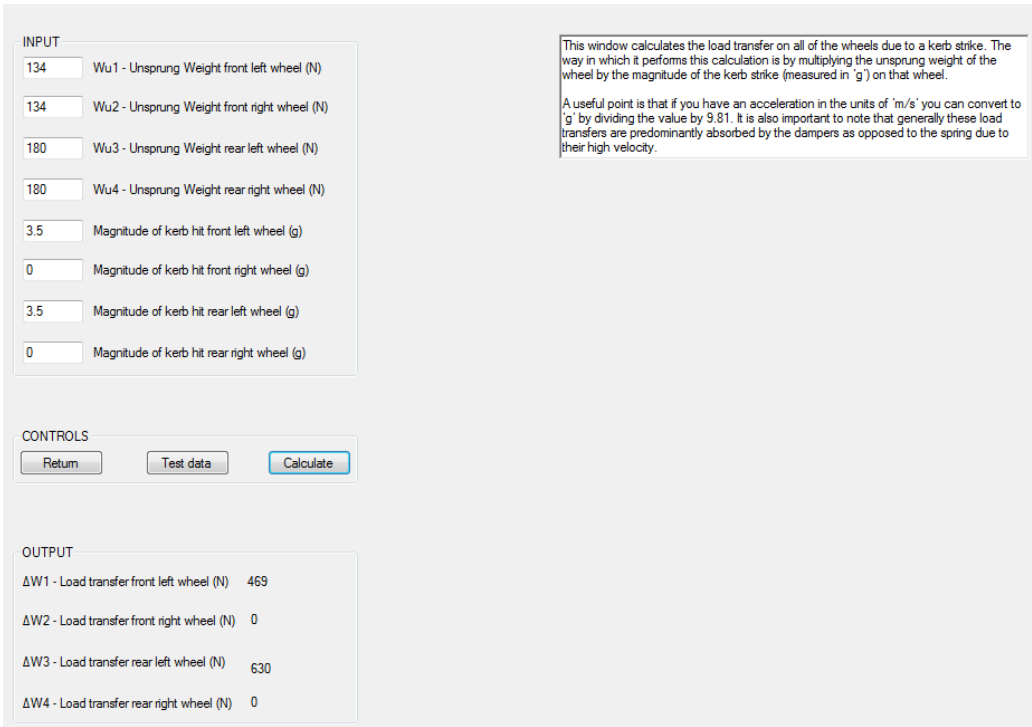


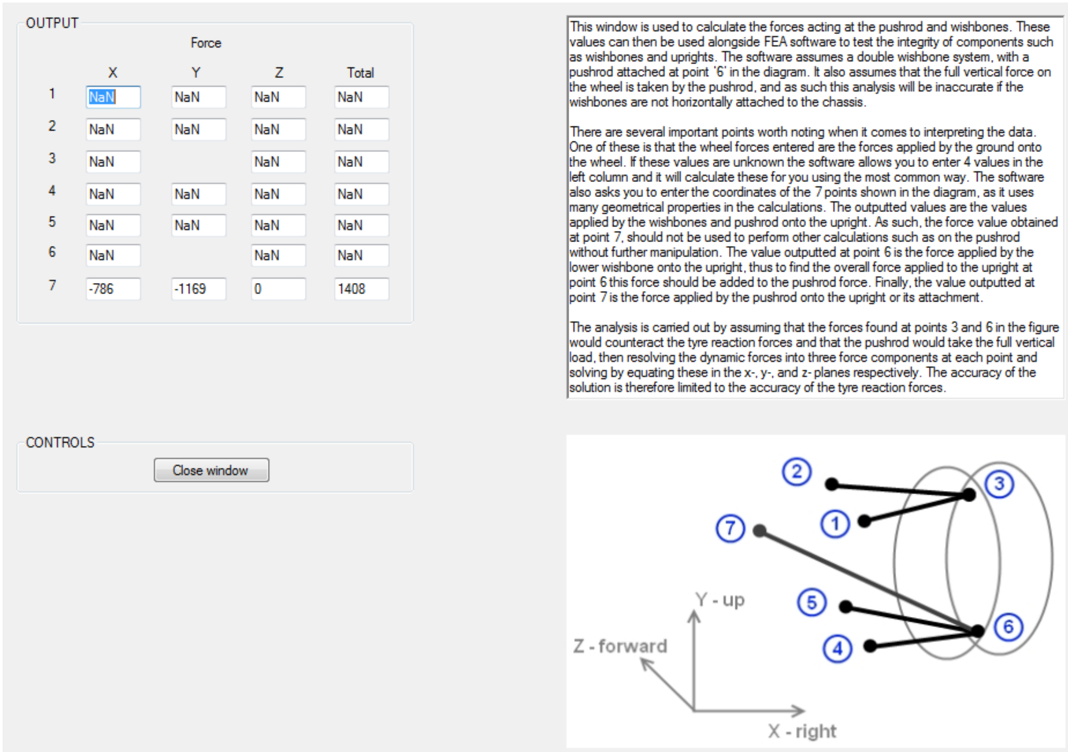
Figure 61: Load Transfer during 1.5g Lateral Load

During a bump event, the affected wheels experience a greatly increased load. To find the total wheel load, the static load must be added to this load transfer. Under a 3.5g bump event, the affected rear wheel experiences a 1330N load and the front experiences 1169N.

$$Wheel\ Load\ =\ Load\ Transfer\ +(Static\ Weight/4)$$

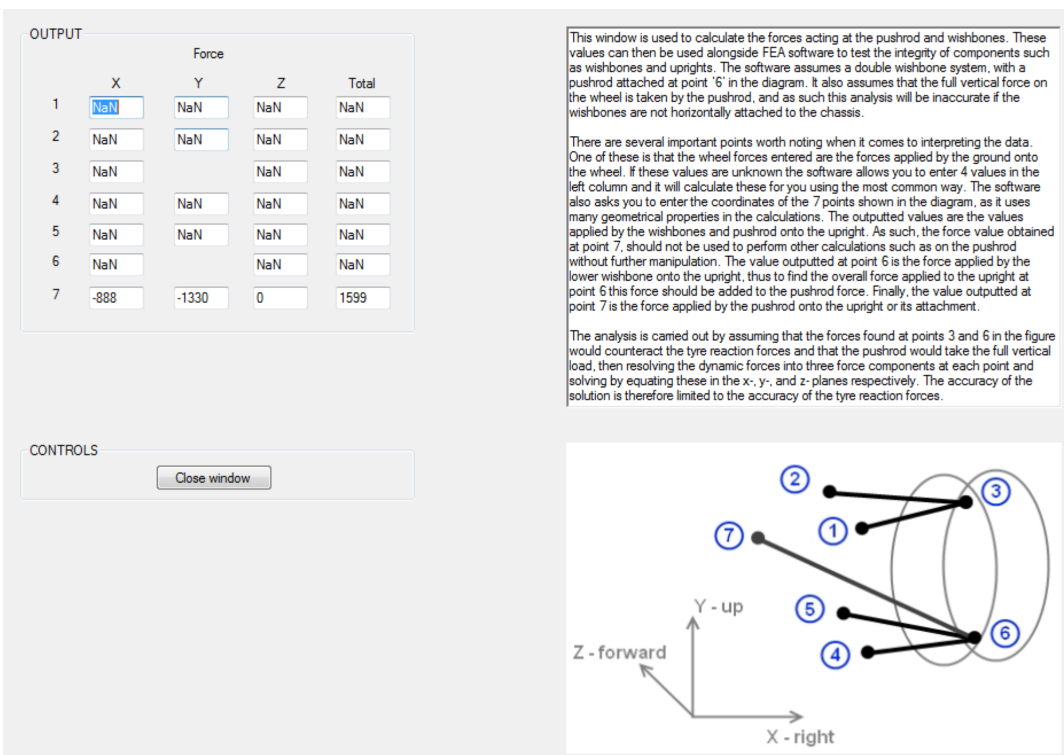
$$1330N\ = 630N\ +\ (2800N/4)$$

$$1169N\ = 469N\ +\ (2800N/4)$$



	X	Y	Z	Total
1	NaN	NaN	NaN	NaN
2	NaN	NaN	NaN	NaN
3	NaN		NaN	NaN
4	NaN	NaN	NaN	NaN
5	NaN	NaN	NaN	NaN
6	NaN		NaN	NaN
7	-786	-1169	0	1408

Figure 62: Front Suspension Arm and Pushrod Forces under 3.5g Lateral Acceleration



	X	Y	Z	Total
1	NaN	NaN	NaN	NaN
2	NaN	NaN	NaN	NaN
3	NaN		NaN	NaN
4	NaN	NaN	NaN	NaN
5	NaN	NaN	NaN	NaN
6	NaN		NaN	NaN
7	-888	-1330	0	1599

Figure 63 Rear Suspension Arm and Pushrod Forces under 3.5g Lateral Acceleration

The 3.5g lateral acceleration forces on the suspension are seen above.

Results Suspension Dynamics

The results from the dynamic suspension calculations show the following.

- Max Pullrod Load:
  - Front 1567N under 1.5g lateral
  - Rear 1599N in 3.5g bump
- Max Control Arm Load:
  - Front 1433N under 1.5g deceleration
  - Rear 492N/496N under 1.5g deceleration and lateral
- Max Wheel Load
  - Front 1301N under 1.5g lateral
  - Rear 1330N under 3.5g bump

SolidWorks Validation

To make sure all simulated results from SolidWorks Simulations were correct, testing was performed using SolidWorks' built in validation tools. These tests were run and compared to actual recorded values in SolidWorks' test files. All simulation results passed without any issues. In addition NAFEMS Benchmarks were also run to further confirm the accuracy of simulations.

FEA Results and Refinement of Final Design

Frame FEA Bump

To make sure the frame meets performance requirements several FEA simulations were run to check the overall deflection, safety factor, and stiffness. Using the highest load conditions found in the suspension dynamic analysis a loading condition is created inside of SolidWorks Simulation. Material properties are 4130 Chromoly Steel Normalized at 870C from the SolidWorks material library.

The first simulation is the front and rear suspension experiencing a twisting load. This would simulate hitting a large bump in the road. The largest value of 1350N (rounded from max front is 1301N and rear is 1330N) was used as a load on a single front and rear wheel.

- Max Wheel Load
  - Front 1301N under 1.5g lateral
  - Rear 1330N under 3.5g bump

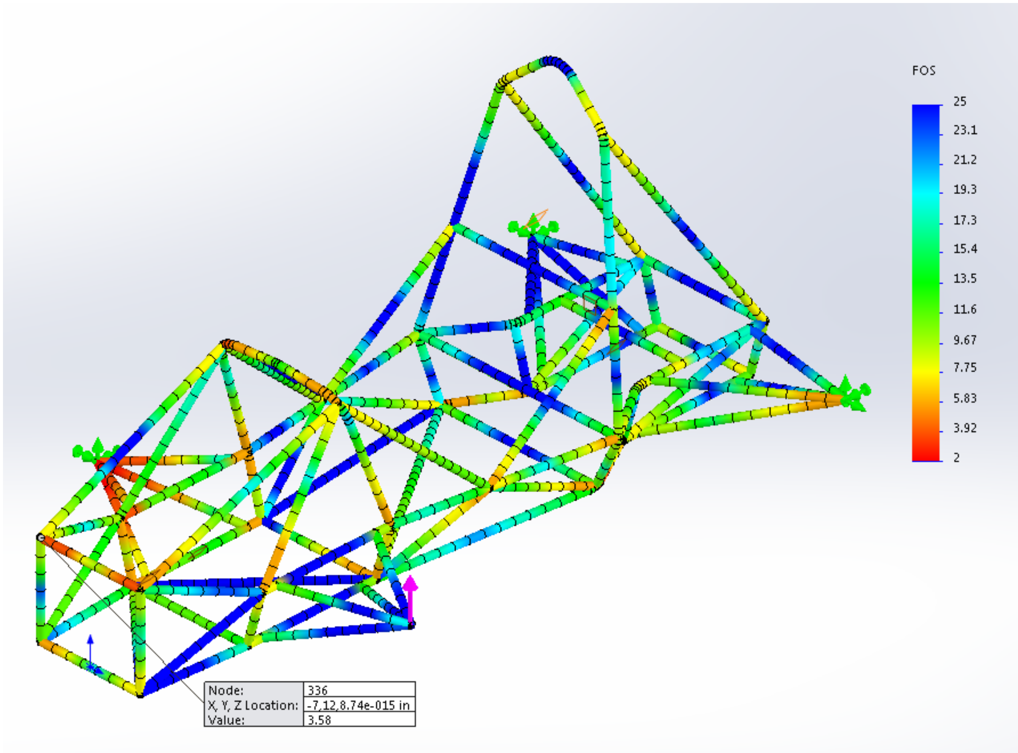


Figure 64 1350N Bump Front Frame Factor of Safety

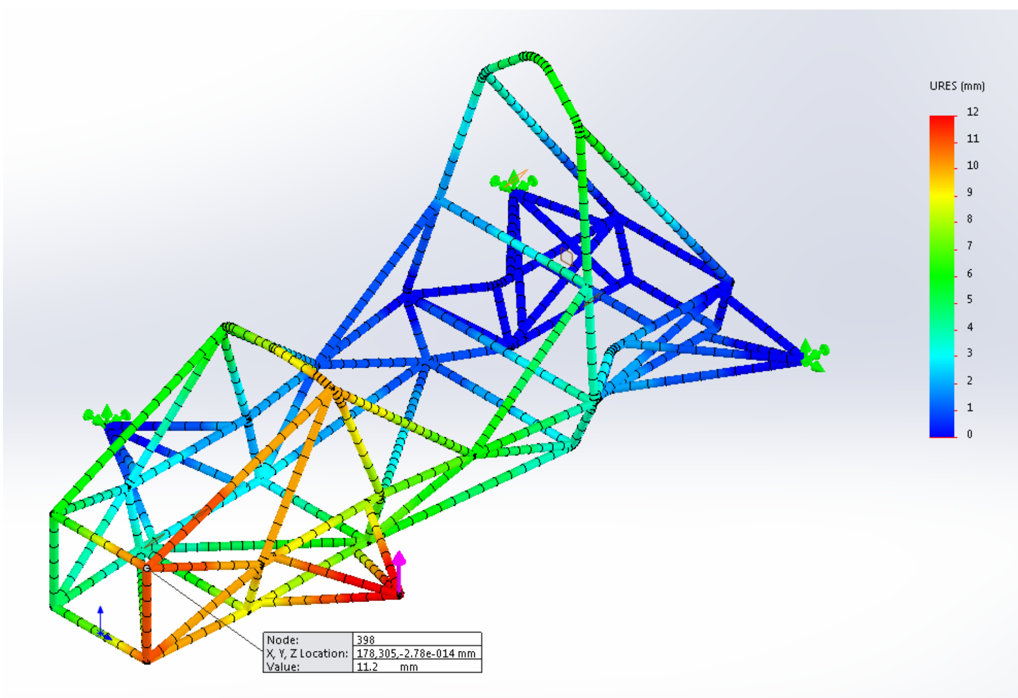


Figure 65 1350N Bump Front Frame Displacement

During a 3.5g impact with front suspension the lowest factor of safety recorded is 3.58 and the greatest displacement on frame is 11.2mm.

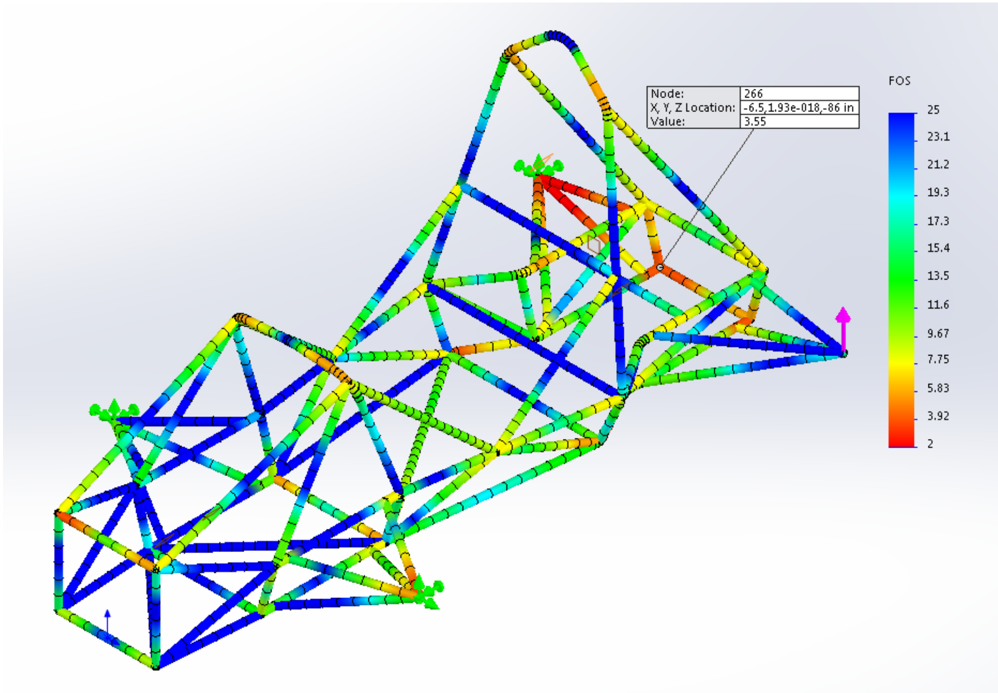


Figure 66 1350N Bump Rear Frame Factor of Safety

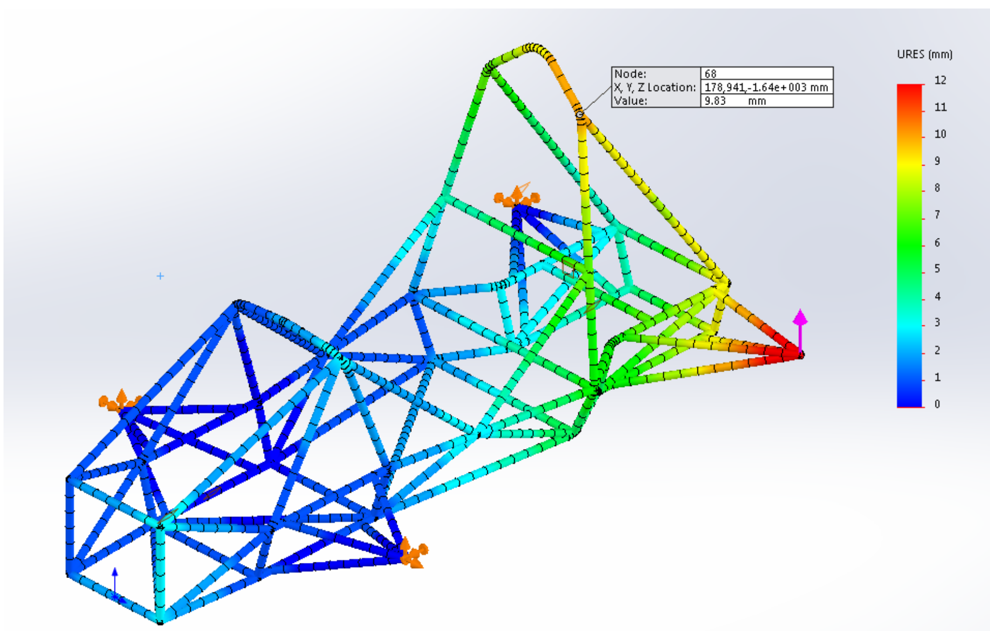


Figure 67 FEA 1350N Bump Rear Frame Displacement

During a 3.5g impact with rear suspension the lowest factor of safety recorded is 3.55 and the greatest displacement on frame is 9.83mm.

FEA Frame Torsional

To quantify the stiffness advantage of the new frame a torsional simulation was run on the new frame and old frame. A 675N force was placed onto the front suspension arms in +/-Y directions. This causes a twisting force on the frame. The rear suspension is fixed in place. The displacement and factor of safety for each frame is shown below.

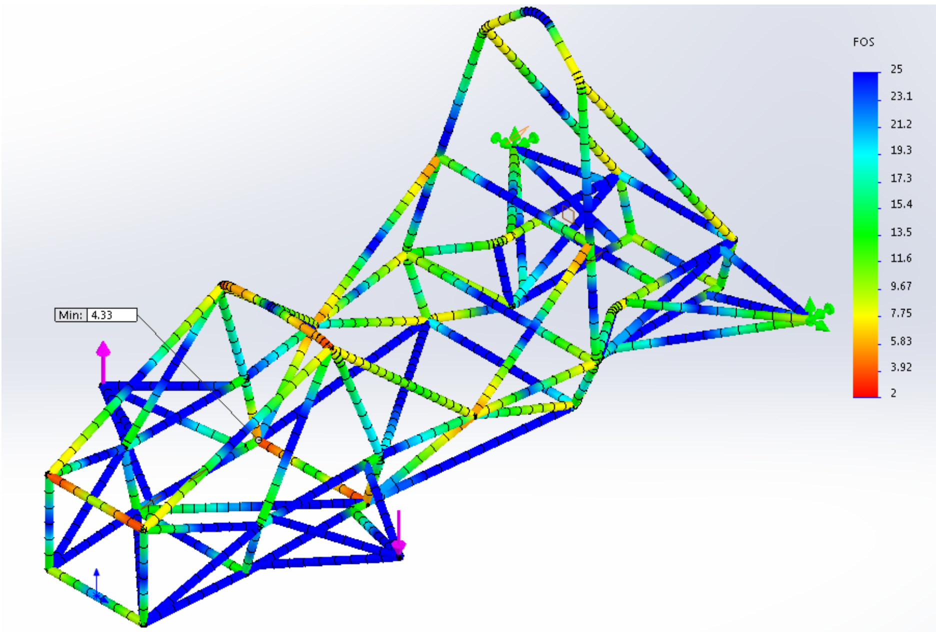


Figure 68 Torsional Load of 675N in +/-Y Factor of Safety

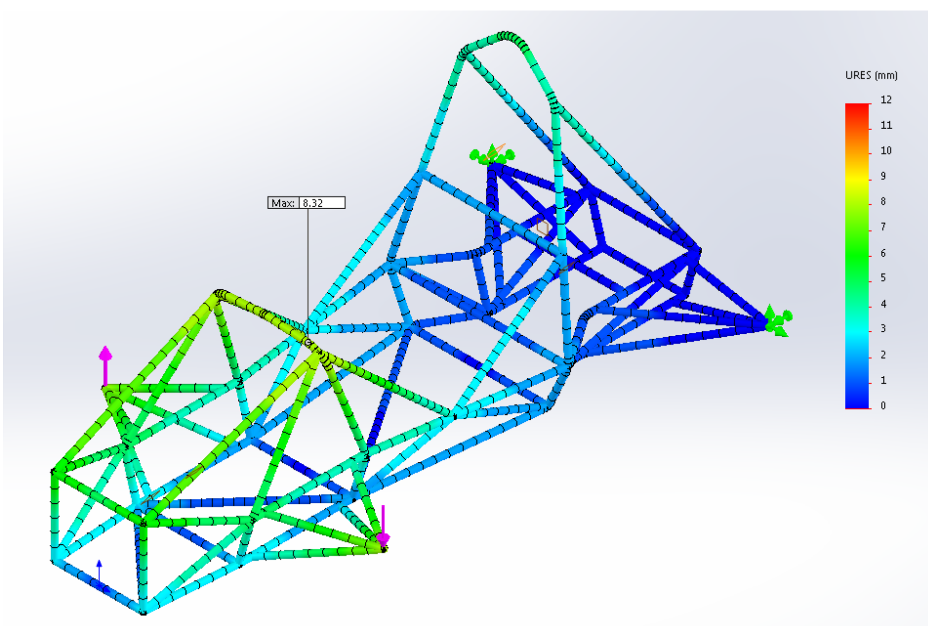


Figure 69 Torsional Load of 675N in +/-Y Displacement

The new frame displaced a maximum of 8.32mm. The lowest factor of safety was 4.33.

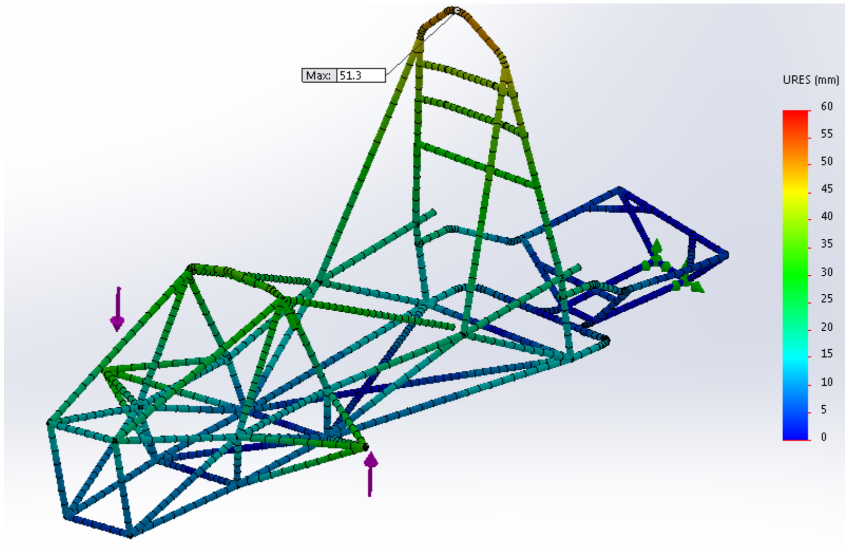


Figure 70 2014 Frame Torsional Comparision Displacement

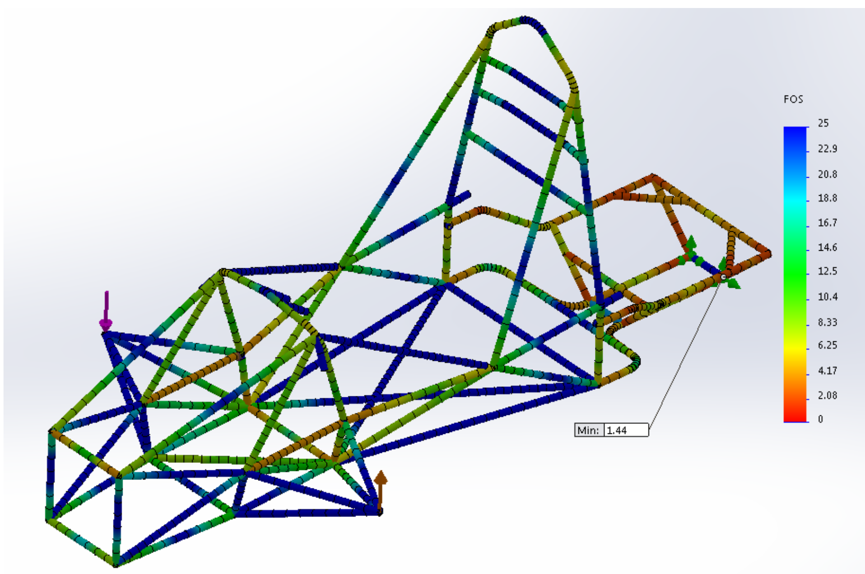


Figure 71 2014 Frame Torsional Comparison Factor of Safety

The old frame with sub frame had a displaced a maximum of 51.3mm. The lowest factor of safety was 1.44.

This makes the new frame around 10 times stiffer than the old frame. Please note that the engine mounting plates, and rear support bars were not included in this test due to modeling difficulty, however the base structure is far stiffer and will offer a great performance advantage to the old frame.

Suspension FEA

To make sure the suspension meets performance requirements several FEA simulations were run to check the overall deflection, and safety factor. Using the highest load conditions found in the suspension dynamic analysis a loading condition is created inside of SolidWorks Simulation. Material properties are 4130 Chromoly Steel Normalized at 870C from the SolidWorks material library.

The simulation is of the front and rear suspension arms experiencing the highest loading conditions. This would simulate hitting a large bump in the road. The largest value of 1600N (rounded from max front is 1567N and rear is 1599N) was used as a load on pullrod mounts. For the lower arms a force of 1350N was used in the X direction and a force of 800N was used in the Z direction. These values are 100N greater than a 1.5g deceleration event.

- Max Pullrod Load:
    - Front 1567N under 1.5g lateral
    - Rear 1599N in 3.5g bump
- Max Control Arm Load:
    - Front 1433N under 1.5g deceleration
    - Rear 492N/496N under 1.5g deceleration and lateral

Front Suspension Arms

For the upper control arms a force of 1600N was placed on the pullrod mounting tab. This simulates a 3.5g bump event. The rear frame mounts are fixed.

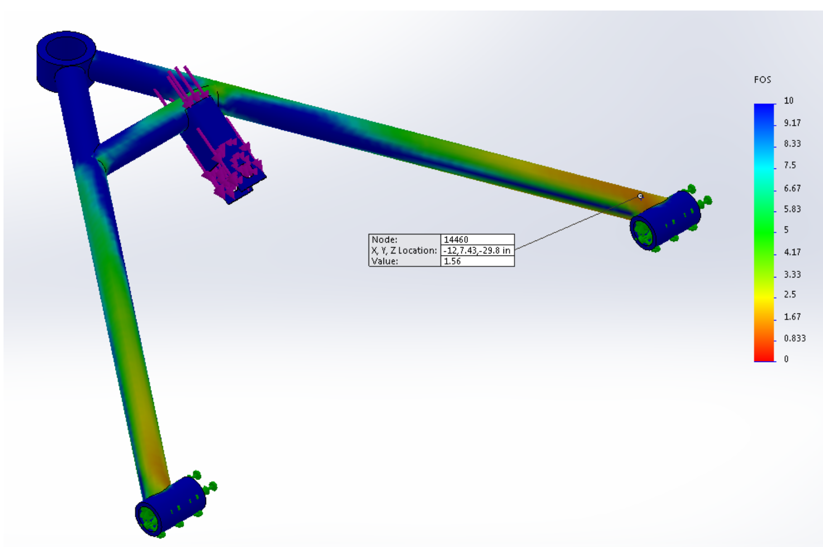


Figure 72 Front Upper Control Arm Factor of Safety

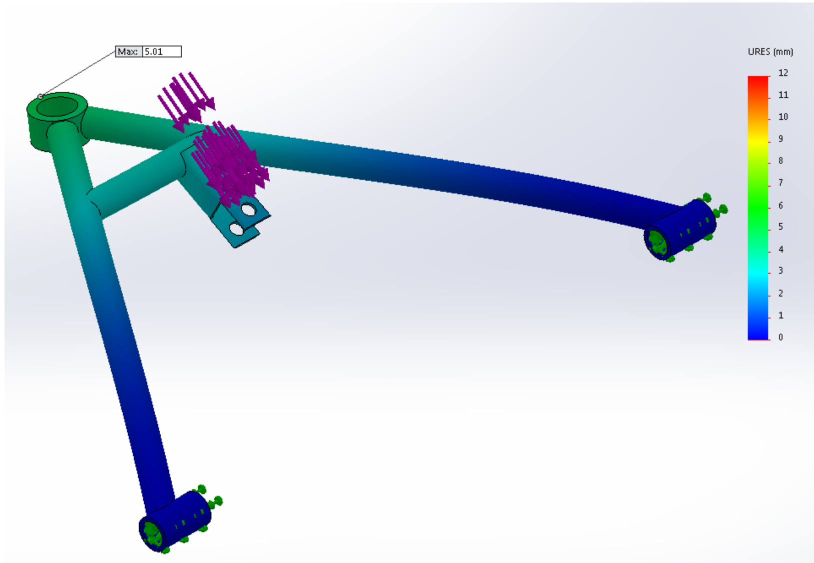


Figure 73 Front Upper Control Arm Displacement

The front upper control arm displaced a maximum of 5.01mm. The lowest factor of safety was 1.56. The factor of safety is a little low compared to other tests; however this was in a sharp corner near the control arm mount.

For the lower control arms a force of 1350N was used in the X direction and a force of 800N was used in the Z direction. This force was placed on the lower spherical mount for the uprights. This simulates a 3.5g bump event. The rear frame mounts are fixed.

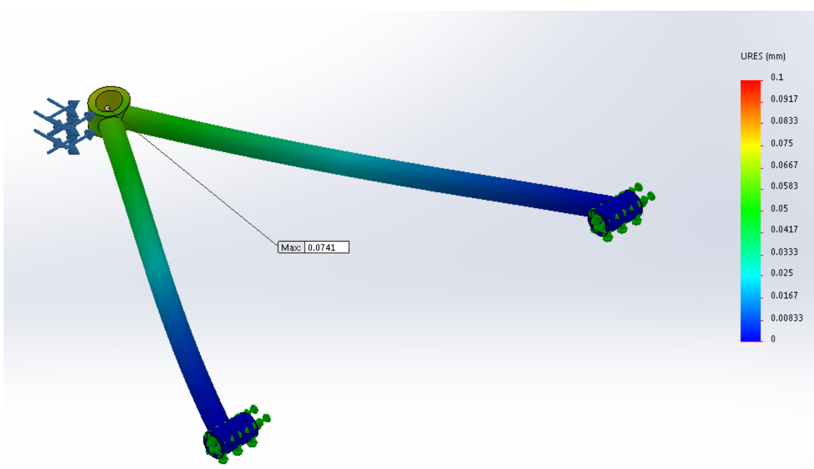


Figure 74 Front Lower Control Arm Displacement

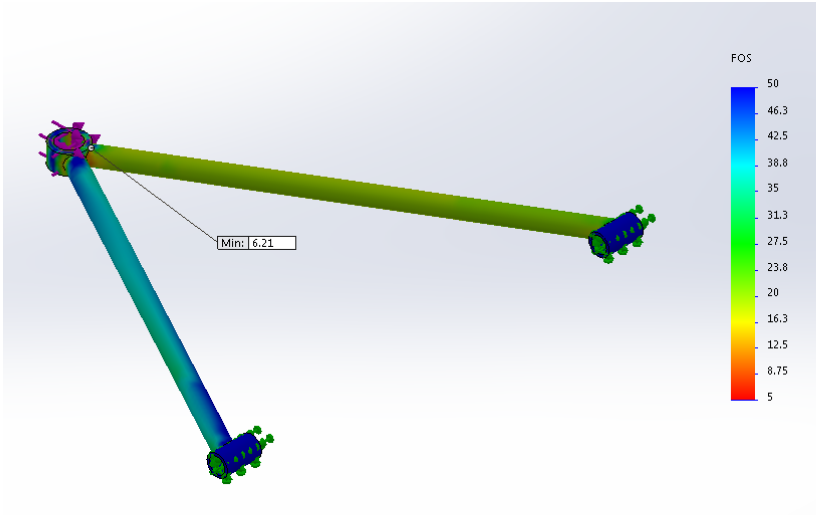


Figure 75 Front Lower Control Arm Factor of Safety

The front lower control arm displaced a maximum of 0.0741mm. The lowest factor of safety was 6.21.

Rear Suspension Arms

For the upper control arms a force of 1600N was placed on the pullrod mounting tab. This simulates a 3.5g bump event. The rear frame mounts are fixed.

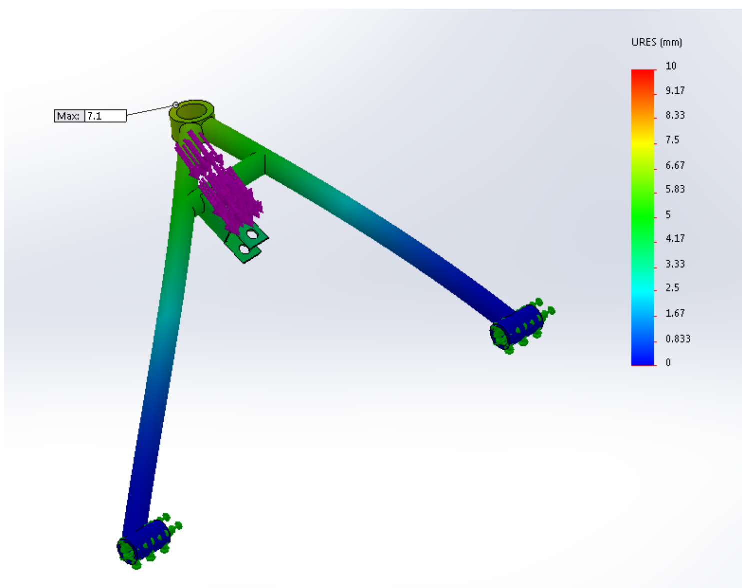


Figure 76 Rear Upper Control Arm Factor of Safety

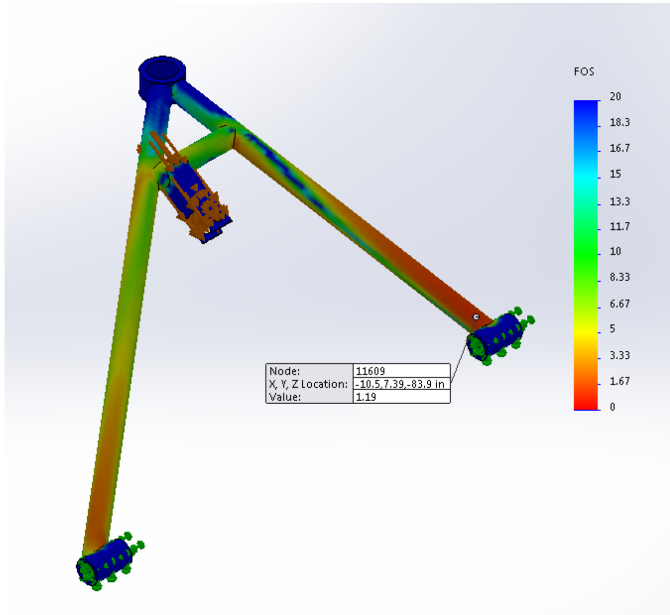


Figure 77 Rear Upper Control Arm Factor of Safety

The front upper control arm displaced a maximum of 7.1mm. The lowest factor of safety was 1.19. The factor of safety is a little low compared to other tests; however this was in a sharp corner near the control arm mount.

For the lower control arms a force of 1350N was used in the X direction and a force of 800N was used in the Z direction. This force was placed on the lower spherical mount for the uprights. This simulates a 3.5g bump event. The rear frame mounts are fixed.

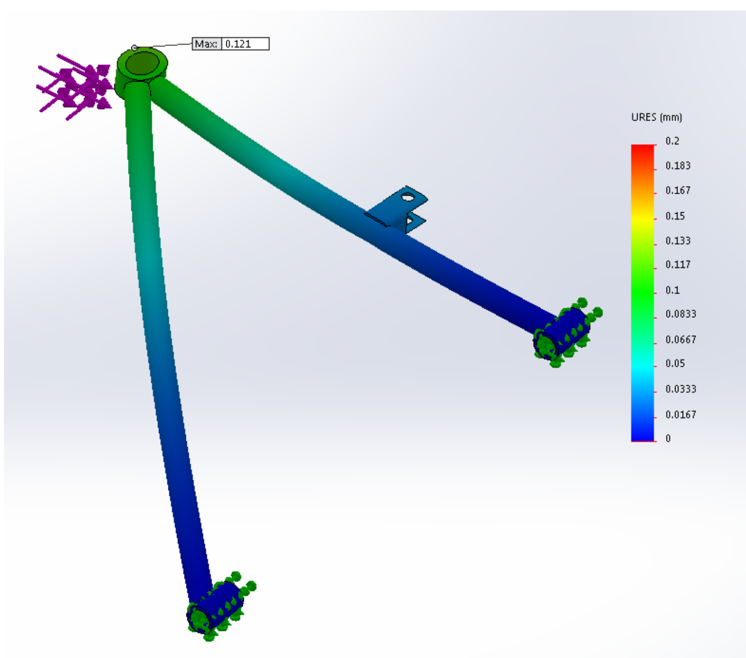


Figure 78 Rear Lower Control Arm Factor of Safety

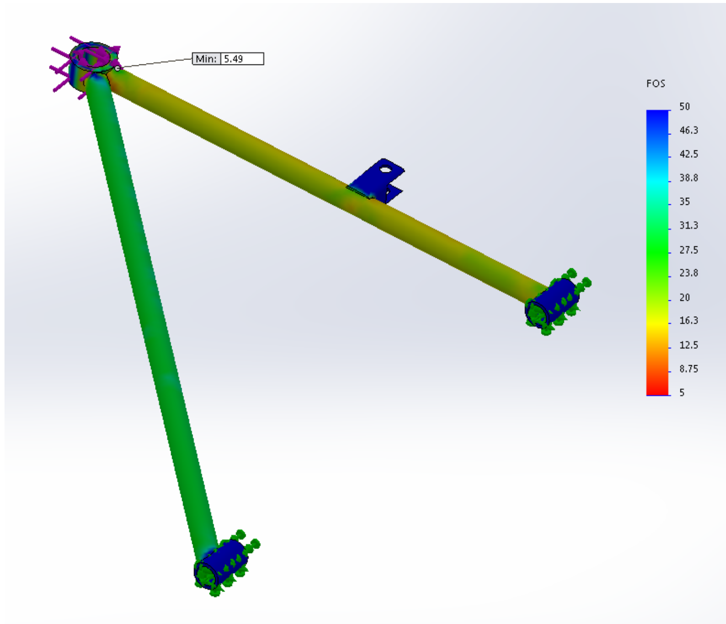


Figure 79 Rear Lower Control Arm Factor of Safety

The rear lower control arm displaced a maximum of 0.121mm. The lowest factor of safety was 5.49.

Manufacturing

Once the design was done the frame was compiled and sent to Cartesian tubing for manufacturing. Cartesian requires 3d geometry of tube profiles to use in their CNC tube bending equipment. These files were prepared and manufacturing took around a week to complete.

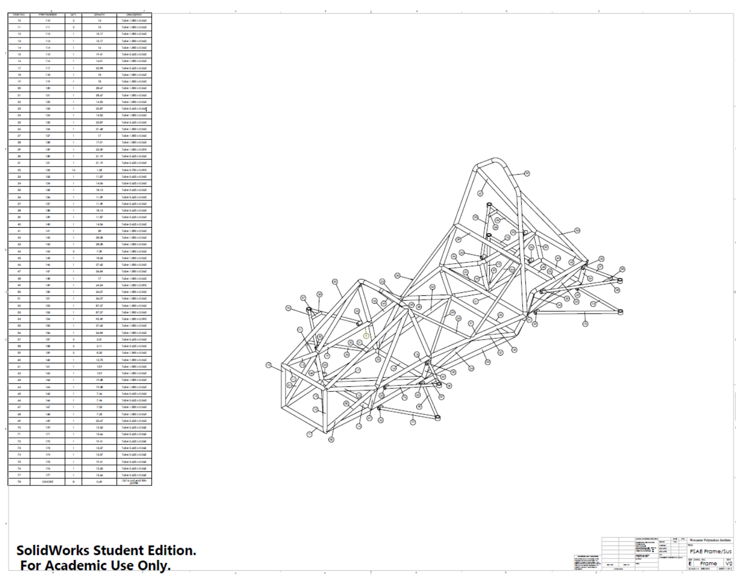


Figure 80 Frame Build File

Once completed the frame was inventoried and setup in the Washburn weld shop. The frame table was used for all fixturing and welding of the frame. This is a heavy duty I beam table made for frame welding by a past MQP.

Figure 81 Fixturing of Front Drivers Cell

Fixturing began by setting up the front bulk head and front roll hoop. Once completed the rear bulk head was tacked in place. This allowed the side impact members and main hoop to be installed.

Figure 82 Main Roll Hoop Installed Free Standing

Once completed and tacked in place the rest of the frame was tacked together. All measurements were doubled checked. The final frame was held to a tolerance of 1/32 of an inch and within 0.5 degrees. The engine successfully fits within the frame as well. The final weight of the welded frame is 64 lbs.

Figure 83 Frame Fully Tacked

Figure 84 Frame Completed with Engine

Conclusion

The final frame and suspension meets all performance specifications set forwards during the design process. The final frame has an overall greater stiffness than the old frame and also weighs less. The larger driver cell provides more room for the driver and the redesigned rear frame allows for easy installation of the engine. The suspension provides great control over camber and roll. The new pullrod damper system provides a greatly improved packaging over direct acting dampers. The pullrod also allows for simple rockers and mounting tabs compared to the rear pushrod system used on the 2014 car. All dynamic situations were considered and the FEA models provided validation that the provided designs will survive in a racing environment.

Brakes, Steering, Uprights & Hubs

Overview of Components

The braking subsystem of the car has perhaps one of the most important functions of all of the components on the vehicle. Stopping the car and passenger is crucial for safety and to control how the car handles during the dynamic competition events. Also, the braking system is evaluated critically during the skid pad event, where it is required that all four of the car's wheels lock at the same time when coming to a stop. Ensuring that this happens takes careful design and fine tuning of the braking components so that they work effectively as a connected system. The braking system is made up of five main components, when the driver inputs a force, it is transmitted directly through the pedal assembly, which magnifies the driver force using simple leverage. Then, this force created is split using a bias or balance bar, which allows for different amounts of force to be directed to the front and rear brakes. The balance bar is connected on either end to a master cylinder, one which controls the front brakes and the other controlling the rear brakes. The master cylinders operate on hydraulic pressure as they are filled with brake fluid fed by a reservoir. Pressure created in the master cylinder is fed through hydraulic lines to the calipers, which physically grip the brake rotors at each of the four wheels. Although this system is closed and has a fairly simple mode of operation, there are many variables to consider when properly designing a braking system.

The steering subsystem is another extremely important component of the vehicle, as it allows the driver to control the path of the car and navigate around the many turns found in the track events of the competition. This subsystem of the car is relatively simple, starting from the steering wheel; the driver input is fed through a steering column which is connected to the steering rack. When the wheel turns, each end of the rack moves either pushing or pulling the tie rods from the center of the car. The tie rods are connected to the uprights, so when they are moved outwards or inwards the wheels will turn.

The uprights function as an extremely important connection between a few systems in the car, and their design affects the suspension, steering and brakes of the vehicle. The uprights physically connect to the tie rods, suspension arms, wheel bearings, and calipers. It is crucial that these connections are precisely placed and supported to ensure proper function of the steering, brakes and suspension.

The hubs of the car are another part of the vehicle that serves as an important connection between various parts and systems. They physically hold the wheel and rotor in place allowing the brakes to slow the wheels. They also are fitted inside the wheel bearings, so they function closely with the uprights as well. The rear hubs of the car also have the function of transmitting the force from the drive axles to the wheels, propelling the car.

Brakes

Major Design Choices

When designing the braking system, there are a variety of different configurations and types of parts that can be used to accomplish the same task. The first major design choice that was considered when creating the braking system was using an inboard versus outboard brake mounting configuration. An inboard brake setup is by far less conventional than an outboard setup, but it offers a few advantages that needed to be considered. In an inboard configuration, the brake rotors are fixed to the axles of the vehicle, typically close to the differential, and the calipers are then mounted to the frame. In an outboard configuration, the rotors are fixed to the hubs in the center of each wheel, and the calipers are then mounted to the uprights.

To make a definitive decision about the mounting location of the braking system a design matrix was used based on various weighted design factors. To fully analyze each system and assist in assessing the value of each category, a list of benefits and drawbacks were created. Based on the table of benefits and drawbacks seen below, it was clear that using an outboard brake setup was a more favorable design mainly due to the ease of access and serviceability of the outboard configuration.

Inboard Brakes		
Decision Factor	Pro	Con
Ease of Access/Serviceability	easier to access hubs and suspension components	difficult to remove rotors, can require removal of suspension
Durability	no need for flexible brake lines, brakes not exposed	more stress placed on axles
Strength of Design	braking forces transmitted to frame or differential	torsional windup in the axles under braking
Simplicity	compact design, central mass	requires more efficient packaging
Heat Management		more difficult to cool
Cost	shorter brake lines, less cost	frame tabs or adapters required
Handling/Performance	Less unsprung mass at wheels	Spongy brake feel due to axle flex
Compatibility	more flexibility in suspension design at uprights	requires custom mounts
Manufacturability	uprights will be simpler	requires mounting tabs on differential or frame

Table 14 Inboard Brakes Decision Factors

Outboard Brakes		
Decision Factor	Pro	Con
Ease of Access	located in very open area of car, easy to access	requires removal of wheel
Durability	much more rigid, shielded by wheels	exposed to track conditions, water, dust,

		debris
Strength of Design	all forces isolated to upright and hub assembly	effects suspension activity
Simplicity	does not interfere with shock mounting or the differential	more complicated line routing
Heat Management	good cooling efficiency, heats tires for more grip	
Cost	will only add small cost to upright machining	flex lines needed
Handling/Performance	great braking feel and stiffness	more unsprung mass at the wheels
Compatibility	currently used on 2014 car, standard in the industry	requires mounting location on uprights and suspension clearance
Manufacturability	does not require any extra components	uprights will need more features and analysis

Table 15 Outboard Brakes Decision Factors

		Concept 1		**Concept 2**	
Decision Factor	**Weight**	**Inboard Mounting**		**Outboard Mounting**	
		Score	Value	Score	Value
Ease of Access/Serviceability	8	5	40	8	64
Durability	9	9	81	7	63
Strength	6	8	48	7	42
Simplicity	8	7	56	8	64
Heat Management	8	6	48	8	64
Cost	4	7	28	8	32
Handling/Performance	10	9	90	7	70
Compatibility	8	7	56	8	64
Manufacturability	7	7	49	8	56
Totals			**496**		**519**

Table 16 Brake Mounting Configuration Design Matrix

When selecting the best calipers for the car, the two major types of caliper configurations had to be considered and the one that would perform best on next year's competition vehicle had to be chosen. The two types of calipers seen in the automotive industry are fixed and floating calipers. Fixed calipers, as their name states, are bolted to the uprights of the car and do not move when operated. The only moving parts a caliper's pistons that push the brake pads inward. Conversely, floating calipers are not rigidly fixed to their mounting location; instead they are mounted to the uprights on slide pins. The caliper body itself will move along the slide pins as the brakes are operated in reaction to the caliper pistons pushing against the brake rotor.

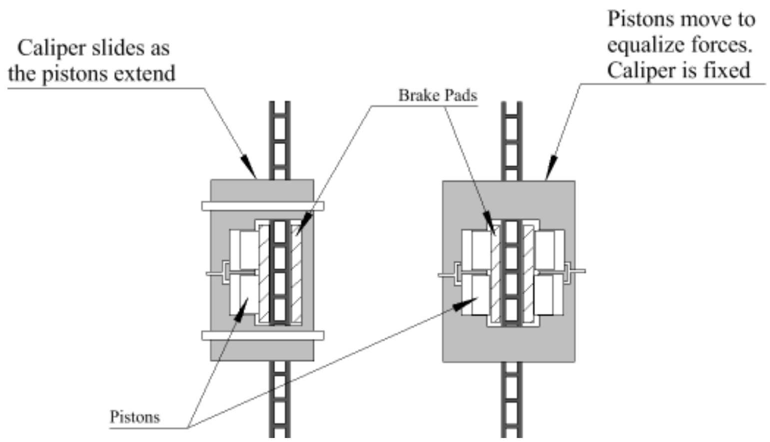


Figure 85 Caliper Types

After developing a comprehensive list of all of the benefits and drawbacks of each type of caliper and adding them to a design matrix, it was decided that using fixed calipers would be a better option for the future car, following the same design as the previous car. The main disadvantage to fixed calipers that was found was the potential for the caliper slides to stick and create large amounts of friction resisting the input from the brake pedal, leading to a poor braking feel. Also, floating brake calipers often have less clamping force than fixed calipers since the force from the brake fluid is only putting pressure directly on one side of the caliper, instead of on both sides. Fixed calipers also are available with multiple pistons which distribute the braking force more evenly and can add large amounts of power.

Decision Factor	Pro	Con
Ease of Access/Serviceability	Service required less often	Service is much more costly
Durability	Better heat dissipation	
Simplicity		More moving parts and hydraulic forces
Braking Feel	Stiffer and more immediate braking feel	
Cost	Small price increase from floating	Slightly more expensive
Stopping Power	Higher stopping power due to higher compression force	
Compatibility	Simple bolt on installation	More difficult to adjust and center rotors properly
Availability	2 available in shop, many used from other teams	

Table 17 Fixed Caliper Design Factors

Floating Caliper		
Decision Factor	Pro	Con
Ease of	Simple service procedure	Needs to be serviced and greased

Access/Serviceability		more often
Durability	Single piston, less sources of failure	Slides are known to sieze
Simplicity	Only one piston used on one side of caliper	Entire caliper moves on pins
Braking Feel		
Cost	Cheaper and more common option	
Stopping Power		Less power than fixed caliper
Compatibility	Self aligning, easier to center rotors in pads	
Availability	Full set available in shop	

Table 18 Floating Caliper Design Matrix

		Concept 1		Concept 2	
Decision Factor	**Weight**	**Floating Caliper**		**Fixed Caliper**	
		Score	Value	Score	Value
Ease of Access/Serviceability	8	7	56	7	56
Durability	8	7	56	7	56
Simplicity	7	9	63	7	49
Braking Feel	7	6	42	8	56
Cost	6	8	48	7	42
Stopping Power	9	6	54	9	81
Compatibility	7	8	56	7	49
Availability	4	8	32	7	28
Totals			**407**		**417**

Table 19 Caliper Design Matrix

Although the rotor is an often overlooked component of a car's braking system; there are many different types of rotors and ways that they are mounted to the car. When deciding on the type of rotors that would work best with the future FSAE car, the decision to use one piece, semi floating, or full floating rotors had to be made. Essentially the difference between these options varied in number of parts and the degree of freedom that each rotor has. A one piece rotor is the simplest design of the three options, it is a single piece of metal that is rigidly bolted to the wheel assembly of the car, and is a fixed part of the assembly. A semi floating rotor is made of two pieces of metal, a central disk and an outer ring that are usually different materials. The two pieces are bolted together and then bolted to the wheel assembly so that in effect the disk is still fixed in place but there are two sections of each rotor to transmit braking forces. A fully floating rotor is almost the same as a semi floating rotor; however, the outer ring is not bolted rigidly to the central carrier. Instead, rotor buttons or float tabs are used to secure the outer ring and carrier together and they allow the outer disk to move laterally a small amount. In this case, the central carrier is fixed to the wheel assembly while the outer ring has a small amount of freedom to move. This may seem like a counterintuitive concept when stiffness and stopping power are crucial but this design offers many benefits and is used on high performance racing vehicles.

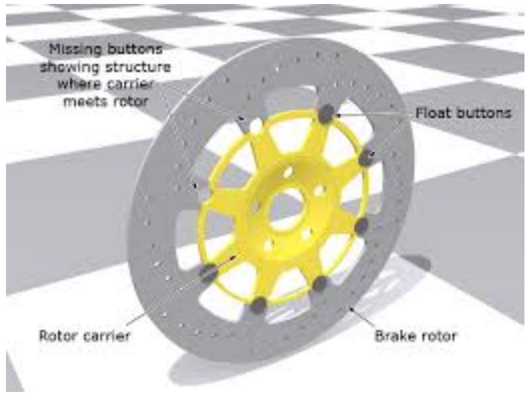


Figure 86 Floating Rotor Design

The two different rotor designs have very different strengths and weaknesses that make each type suitable for different applications. The single piece solid rotor is much cheaper and easier to install, since it does not have the extra float tabs and center carrier. For this reason it is often used in applications where cost a top priority and top level performance is not necessary. The major issues with this design of rotor is that under high use situations the rotor can heat up and expand, causing the metal to warp and deform. This effect is not a problem for floating and semi floating rotors, since the heat stays in the outer ring and is not transmitted to the carrier. This is one of the main advantages of the floating rotor, and because the outer disk is separate from the center the rotors will not warp and will not fade when heated. Another important advantage of floating rotors is that they have a small amount of lateral freedom due to the float tabs creating a non-rigid connection between the two pieces. This allows for the disk to self-align within the caliper, in case the hub and upright do not line up perfectly. This eliminates any rubbing or bending stress on the disk in case it is slightly off center in the caliper. This is one of the main reasons why a floating rotor was chosen, because when using a fixed style caliper that cannot adjust itself to the disk, room for error and adjustability is needed. Also, in our case the hubs will be used as the center carrier for the rotors, which lower the overall cost of the rotors and let us choose from a wide range of products on the market. As you can see below in the design matrix, floating rotors proved to be a better choice for our car according to our analysis.

		Concept 1		**Concept 2**	
Decision Factor	**Weight**	**1 Piece**		**Floating**	
		Score	Value	Score	Value
Ease of Access/Serviceability	6	7	42	7	42
Durability	8	6	48	8	64
Strength	8	8	64	7	56
Simplicity	7	8	56	7	49

Heat Management	9	6	54	9	81
Cost	5	8	40	6	30
Stopping Performance	9	7	63	9	81
Compatibility	7	6	42	9	63
Manufacturability	7	7	49	6	42
Totals			**458**		**508**

Table 20 Rotor Design Matrix

Although rotors come in two main categories, there are also a number of finishing features done to the braking surface to increase performance. Although many automotive rotors are solid machined disks, most performance rotors are either drilled, slotted or both. Drilled rotors have holes drilled completely through the rotor to decrease weight and allow for air to flow through the braking surfaces. The holes in the rotor greatly improve its cooling ability, and allow for brake dust and debris to be expelled much easier instead of being clamped on the surface of the disk. Also, almost all motorcycle and small vehicle rotors are drilled from the supplier, so it would be very easy for the team to find an off the shelf product that meets our requirements. The only possible issue with drilled rotors is that the holes can develop cracks over time due to heat cycling and can slightly weaken the rotor. Slotted rotors have radial grooves or slots machined in the braking surface of the rotor that do not go entirely through the rotor itself. These rotors are very effective at clearing brake dust from the disk surface and maintain the original strength of the rotor before machining. However, the weight savings of this operation is minimal and these rotors are much more costly.

		Concept 1		**Concept 2**	
Decision Factor	**Weight**	**Drilled Rotors**		**Non-Drilled Rotors**	
		Score	Value	Score	Value
Durability	7	6	42	7	49
Strength	7	6	42	8	56
Simplicity	8	7	56	9	72
Heat Management	9	10	90	5	45
Cost	5	7	35	8	40
Stopping Performance	9	9	81	7	63
Weight	7	9	63	6	42
Manufacturability	6	7	42	9	54
Totals			**451**		**421**

Table 21 Drilled Rotor Design Matrix

		Concept 1		**Concept 2**	
Decision Factor	**Weight**	**Slotted Rotors**		**Non-Slotted Rotors**	
		Score	Value	Score	Value
Durability	7	8	56	7	49
Strength	7	8	56	8	56
Simplicity	8	7	56	9	72

Heat Management	9	9	81	5	45
Cost	5	7	35	8	40
Stopping Performance	9	9	81	7	63
Weight	7	7	49	6	42
Manufacturability	6	7	42	9	54
Totals			**456**		**421**

Table 22 Slotted Rotor Design Matrix

As shown in the design matrices above, it was found that it would be beneficial to use rotors that have been both drilled and slotted. However the decision was made to use only drilled rotors for the 2015 car. Drilled rotors were not selected because the weight savings of drilled rotors is significantly greater and having both holes and slots on such a small rotor will not increase the performance by much. Since there is a limited surface area on the rotors it would be difficult and costly to use rotors that have been both drilled and slotted. Also, the team has used drilled rotors with much success in the past and there are very few suppliers that make slotted rotors that will work for the FSAE application. After comparing many different suppliers and rotor options, it was chosen to use the 220mm rotor with a thickness of 4mm made by RCV Performance. They have a significant supply of FSAE components and have years of experience designing parts and working with the Drexler FSAE team.

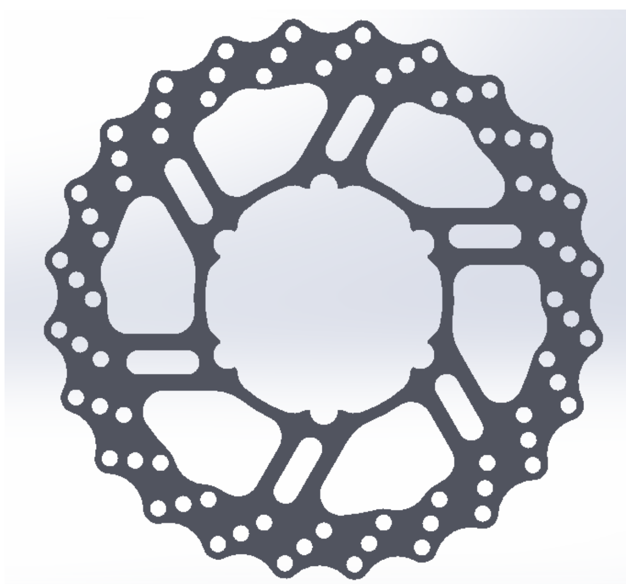


Figure 87 Solidworks Model of RCV Rotor

Calculations

In order to assist in selecting the best components for the braking system and to ensure that the car will stop when needed, detailed calculations had to be made to analyze the forces produced during hard braking situations. The first objective of the force calculations was to develop a requirement for the components being designed and to evaluate the dynamic shift in weight during braking so that a front and rear baseline bias can be found and followed. First, using a general predicted weight of the car and driver along with typical deceleration values seen during dynamic events, the total force required to stop the car was found. Using this value combined with an estimate of the center of gravity based on last year's car and a 50-50 weight balance, it can be seen how the car's weight is transferred to the front of the vehicle. Using weight transfer equations the force that is shifted to the front wheels was isolated and the exact force that must be transmitted to each wheel in a worst case scenario to stop the car was found. Once these requirements of the system were known, the components can now be modeled and optimized in a series of calculations.

From resources such as stoptech.com and Wilwood, a live excel spreadsheet was created that allows us to easily see the effect of using various components in the system. It also allows for the bias, driver input force, and many other variables to be edited if needed. The system equations follow the path of the input force of the driver through the pedal and each of the components in the system, eventually ending at the wheels. The calculations are split between the front and rear wheels, since the force from the driver is immediately split by the bias bar at the pedal assembly. This way, different components can be used for the front and rear systems to ensure that the proportion found earlier is met. Another benefit of this method is that the front to rear brake proportion will be designed into the system hardware, and can be fine-tuned using the balance bar if necessary. The final parameters of the braking system calculations are included in the appendix of this report with the excel document used.

Component Selection

Using the braking system design calculations, the exact products that would be purchased and used together for the 2015 vehicle were confidently selected. Our team focused on compatibility, cost, availability and weight when selecting components that met the needs found by our system simulation. For the pedal assembly, the same Wilwood brand brake pedal assembly as the 2014 car was chosen for many reasons. First, the pedal has the typical 6 to 1 ratio that provides plenty of force for unassisted hydraulic brakes and it has already been proven during our testing. Also, the Wilwood pedal assembly is designed to house two separate master cylinders which are required by the rules of the competition. This allows us to choose different size master cylinders if needed and uses a brake balance bar to further tune the front to rear braking bias. For master cylinders, since the team was already using the Wilwood pedal assembly, it was decided to also use Wilwood master cylinders since they will mount to the pedal with no issue and are proven through use in the current car. They also are small and lightweight, using a reservoir that can be relocated if necessary to save space.

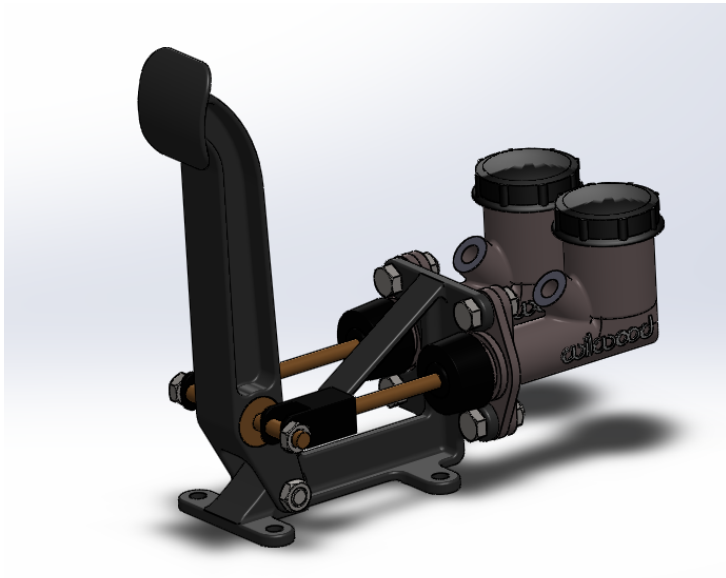


Figure 88 Solidworks Model of Pedal and Master Cylinder Assembly

Continuing with the use of Wilwood products, the team chose to use Wilwood PS-1 model calipers for the rear brakes and Wilwood Dynalite calipers for the front brakes. The Wilwood PS-1 brake calipers are a smaller, very lightweight caliper that is perfect as a rear brake where not as much force is required. Although our team has not used them in the past they have been proven by many other teams and are very similar to the Brembo brakes used in the 2014 vehicle. The Dynalite calipers are a larger, more powerful caliper that will work well as a front brake, where most of the braking forces are transmitted. This caliper is being used on the 2014 vehicle with no issue and there are some available in the FSAE shop already, so they will not need to be purchased.

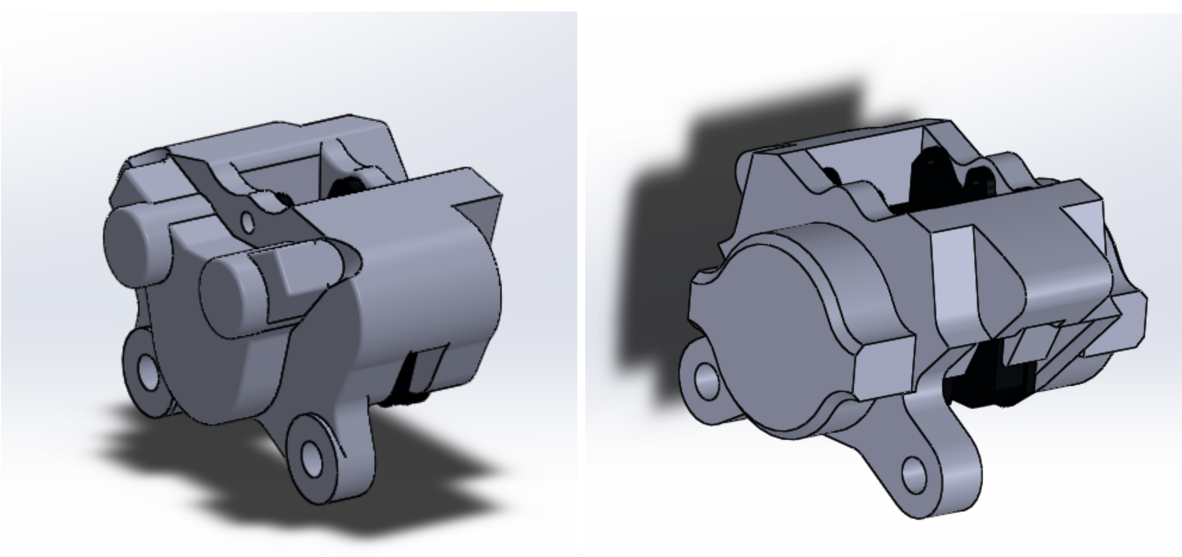


Figure 89 Wilwood PS-1 and Dynalite Calipers

The rotors for the car were much more difficult to source. Unlike some of the other components, rotors are often made by teams since products are very model specific. It was considered to design and cut our own rotors; however using a readily available product that is tested and made of the correct materials was more favorable. The rotors chosen are floating rotors made by RCV performance, which is a reputable brand within the FSAE community. The 220mm rotor of 4mm thickness will fit well within our wheel well and is both drilled and floating as was desired. These rotors are very cheap and easy to replace if necessary during the life of the car. As a whole, the components chosen for the braking system will be extremely reliable and powerful.

Steering

Major Design Choices

The steering components of the FSAE vehicle are a crucial system that must be designed for ultimate reliability and safety. For the design of the system there were two main decisions that needed to be made by the team. First, the steering rack location had to be chosen and then the steering rack itself had to be selected. When deciding the location and configuration of the steering rack, there were two common options that were compared. The first option, which is less common, would be to mount the steering rack above the driver's shins in the cockpit. The advantage of this placement is that a solid, single piece steering shaft can be used, leading to more responsive steering and better overall feel. This option also lowers the weight of the steering system, since the rack is closer to the wheel. However, this option has considerable faults. Raising the steering rack raises the overall center of gravity of the car, which will increase body roll forces and make cornering more difficult. This setup also poses the risk of injury to the driver in an accident, in which case their shins would impact the steering rack directly. Having the rack placed in this area of the cockpit also makes it difficult for the driver to escape the vehicle in the event of a fire, which is a critical safety hazard. The second and more common option for steering rack location is to mount it to the floor of the cockpit beneath the driver's legs and out of the way of their body. This design maintains a low center of gravity and is more comfortable and safe for the driver. Unfortunately, this configuration requires a two-piece steering shaft with a U-Joint connection, adding complications to fabrication.

The team chose to mount the steering rack on the floor of the vehicle, due to the safety concerns associated with an elevated mounting position and mainly due to the success of this configuration with the 2014 car. Even though this design requires an extra steering shaft and U-Joint, these components are very easy to procure and manufacture, and do not have high costs. Also, jointed steering shafts are extremely common in full size cars, and when assembled correctly will be just as durable as a solid shaft.

Knowing the hardware that will be used to construct the steering shaft and tie rods, now a steering rack must be chosen from the various suppliers available. There were four major brands that made steering racks for the FSAE application; Formula Seven, KAZ technologies, Stilletto, and mRack. The main differences between these steering racks were weight and cost. As seen below, when assessing the practicality of each product, it was found that the most affordable option, the Stiletto rack, would be best. This steering rack had a relatively low weight compared to its more expensive competitors and was the simplest of the four options. In a design competition where cost is a judging factor, the small advantages of the other steering racks could not be justified. Other teams have proven the reliability and strength of this steering rack, and it is very similar to the one used on the 2014 car.

		Concept 1		Concept 2		Concept 3		Concept 4	
		Kaz Technologies		Formula Seven		mRack		Stiletto	
Decision Factor	**Weight**	Score	Value	Score	Value	Score	Value	Score	Value
Serviceability	5	7	35	6	30	6	30	8	40
Durability	8	8	64	7	56	9	72	7	56
Simplicity	8	7	56	6	48	6	48	9	72
Weight	7	6	42	7	49	10	70	6	42
Cost	8	7	56	5	40	5	40	10	80
Size	5	7	35	7	35	8	40	8	40
Steering range	9	8	72	7	63	6	54	9	81
Totals			**360**		**321**		**354**		**411**

Table 23 Steering Rack Design Matrix

The Ackerman Steering Principle

When designing a steering system, the most important factor to consider is the level of Ackerman used in the assembly geometry. The Ackerman principle is based on the fact that when a car is turning perfectly around a corner with no slippage, the inside wheel will track on a circle of a different radius than the outside wheel, and therefore each front tire will be at a slightly different angle. The geometry of this condition is designed so that all four wheels of the vehicle pivot around a singular point and do not oppose each other in any way while rolling. This is shown in the figure below.

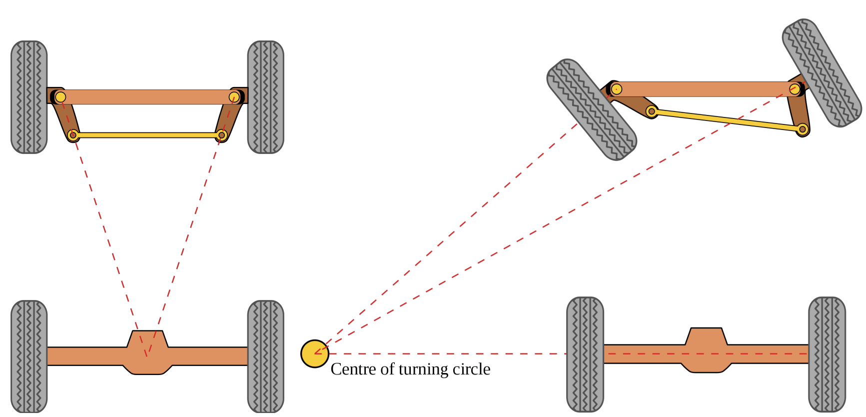


Figure 90 Ackerman Steering Geometry

As you can see, the angle from the center of the turning circle to each wheel is different, with the inside wheel having the larger angle to the rear axle. This setup is considered to be 100% Ackerman, since it creates no slippage in on the front wheels while the car is turning. This setup may seem ideal and make sense for all driving conditions, but in reality hardly any vehicles made today incorporate this design. On typical FSAE vehicles, the steering systems incorporate positive Ackerman, which means that the imaginary line drawn from the tie rod connections to the rear axle intersect either on or in front of the rear axle. This will cause the inside tire to lead the car into the turn, as it will react more quickly than the outside wheel. Using a positive Ackerman setup is beneficial for driving at slower speeds on tracks with tight turns such as the endurance course at the national competition. For this reason it was decided to use positive Ackerman in the setup of the steering components. The mounting points for the tie rods on the uprights are designed to be closer to the center of the car than the suspension joints, creating the Ackerman geometry. As shown below from the SolidWorks model, the lines from the uprights intersect in front of the rear axle of the car.

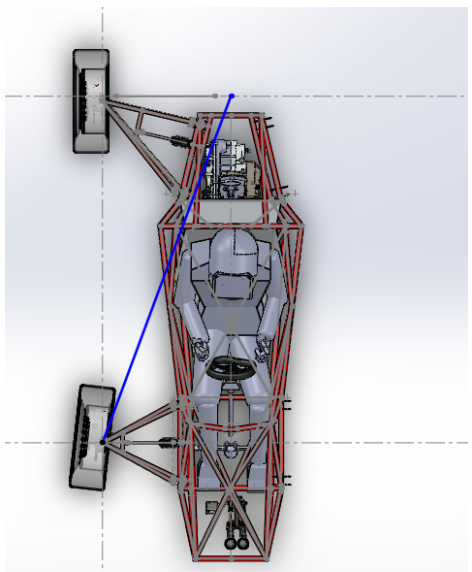


Figure 91 Ackerman Measurement

It is important to note that the position of the steering rack will affect the steering behavior. For example, if the rack is moved away from the driver, the car will have less Ackerman and when it is moved closer it will induce more Ackerman. This concept can benefit the tuning phase of the vehicle's development, because the rack could be easily moved to test different steering conditions. This is one of the main reasons why the exact position of the steering wheel and rack were not chosen by our team. Since the steering wheel and fabrication of steering components will be done by the next FSAE team, the exact details and dimensions of the connecting parts were left up to their decision. This way, they can ensure that their driver is comfortable in the seat of the car and their wheel interfaces correctly with the frame and other parts.

Hubs

Previous Design

The hubs that were used on the 2014 FSAE vehicle were designed to be machined from standard 6061 Aluminum stock on the WPI automated lathes and mills. As seen in the figure below, the design concept is straightforward. The hub mates to three separate components within the wheel assembly; the disk, the wheel and the wheel bearing. The outer face matches the mounting hole locations for the wheels used in the common 4x100mm spacing layout. The rotor mounting location is behind the wheel mounting tabs, and is designed to line up perfectly with the rotor mounting points and float tabs used to hold the rotor on the hub. These mounting tabs must be correctly spaced from the wheel mounting tabs to ensure that the rotor will be directly centered in the caliper body and not interfere with the upright as well. Behind the rotor mounting tabs is the spindle, which interfaces with the wheel bearing. This shaft must have the same diameter as the inside of the wheel bearing to ensure there is no shaft play between the two parts. On the spindle of the front hub, there must be a fastening feature that prevents the hub spindle from moving laterally inside the wheel bearing. On the rear hub this is not needed because the axles restrict lateral movement and prevent the hubs from moving away from the car's center plane. On the previous design, the front hubs used a back plate that is fastened to the spindle using a standard bolt. This design works, however it significantly improved. The major issue with this setup is that the backing plate is much larger than needed, so it contacts the upright, causing unnecessary wear and noise. Since the backing plate rotates with the hub, it actually causes excess friction and heat during driving. It is also important to note that the hubs have hollow centers, both to reduce weight and to house the axles inside the rear hubs.

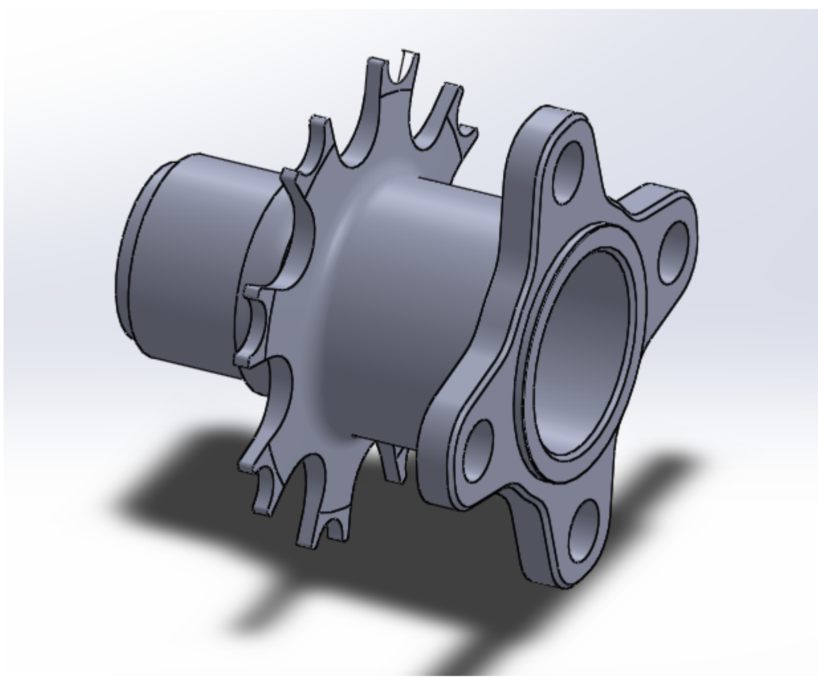


Figure 92 Previous Hub Design

Design

To successfully redesign the hubs of the vehicle, changes needed to be made to accommodate for new rotors, calipers and a better front spindle design. First, to ensure that the rotors will mount to the hubs, the mounting tabs needed to be shortened and matched to the dimensions of the rotor. As seen below, the tabs are much closer to the center axis of the hub, which has the positive effect of decreasing weight and increasing their strength. Once the rotor could be properly assembled to the hub, the location of the mounting tabs in relation to the wheel tabs was refined. By moving the mounting tabs laterally on the hub body, the rotor can be placed in the center of the caliper and a proper mounting location for the caliper can be found. Using an assembly model shown in the side view below, the location of the mounting tabs was designed to insure no interference between the wheel and caliper on both the front and rear hubs.

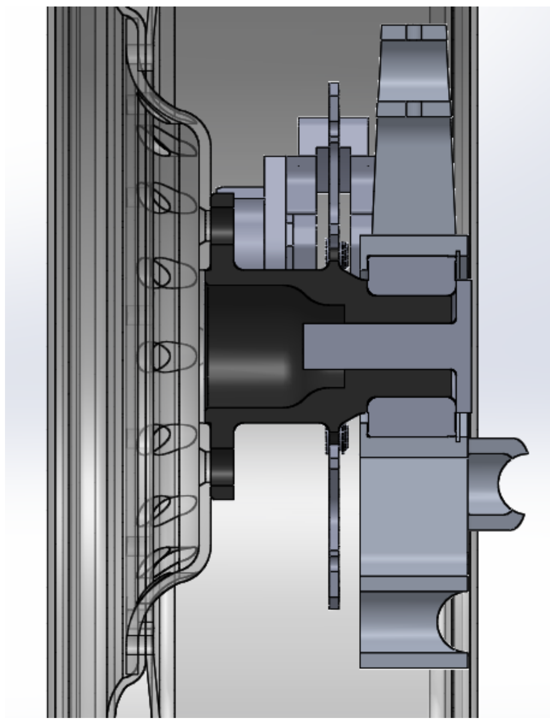


Figure 93 Section View Interference Check for Rear Hub Assembly

Using this dimension, the mounting tabs for the calipers now are easily incorporated into the upright design to ensure proper location of the caliper. Next, a new fixturing system was designed for the front hubs that would not cause interference between the wheel assembly components. Using inspiration from hub designs used by other FSAE teams and suppliers, the team decided to use a deadaxle and locknut as seen above. This is a simple design that will be easy to assemble and reduces the number of parts in the overall assembly. Also, it ensures that the hub and upright will not interfere while the wheels are rotating.

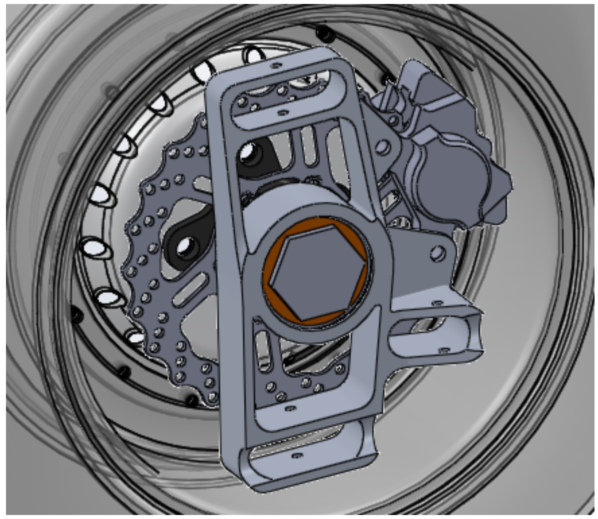


Figure 94 Front Hub Design with Deadaxle

On the rear hubs, the past design used a matched insert to mate the axle spline to the hub. This system has proven to be effective for our team but the option of incorporating the spline into the rear hub and eliminating the need for an insert is also available. However, after comparing the two options the team decided that using an internally splined hub would be a better option for the future car. By machining a spline into the internal bore of the hub, it allows the hub to be made from one single piece of aluminum instead of using a steel insert. This lowers the overall weight of the hubs and makes the assembly process easier. Also, this way inserts will not need to be cut from ATV hubs to match the inner pattern of the hubs.

Each set of hubs will require wheel bearings to ensure that all of the wheels rotate without restriction and can withstand driving forces on an autocross course. For the front wheel bearings, the team has decided to continue using BMW E30 rear wheel bearings since they have worked very well on the current car and they are inexpensive to obtain. These bearings are quite large and can handle the torque of the front wheels turning into and out of corners on the track. For the rear wheel bearings the team has selected to use the stock front wheel bearing from a Honda TRX 500 ATV. Using these wheel bearings will allow our axles to properly mate with the bearing faces as they do on the ATV, and they are easy to source. Also, they are smaller and lighter than the bearings on the current car, which allows the rear uprights and hubs to be smaller and lighter as well.

Validation

To ensure that the designed hubs will survive the driving conditions seen in competition, it was critical that finite element analysis simulations were conducted using the 3D models in Solidworks. For the front and rear hubs, both extreme braking and bump scenarios were tested. The spindle of each hub was fixed to simulate the connection between the wheel bearing and hub. Then, four point loads were added to the wheel stud mounting points to either simulate a braking torque or an upward impact from rolling over a bump in the track. For the braking test, the rotor mounting points were also fixed to simulate when the wheels have locked and the vehicle is sliding. For the braking tests, both hubs had a factor of safety (FOS) of over 2.5, and for the bump tests the FOS was over 5. This is a very good factor of safety for these components and is an improvement over the strength of the current car’s components.

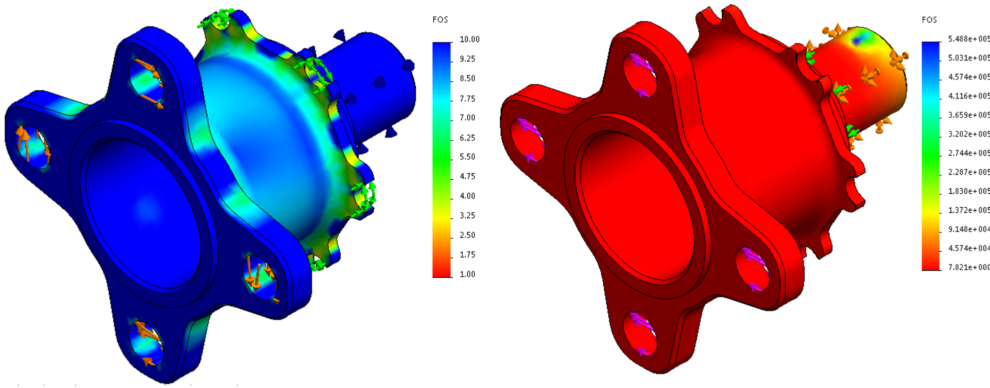


Figure 95 Factor of Safety Plots for Front Hub

Uprights

Design

The uprights are an extremely important part of any vehicle, as they connect the wheel, brakes, steering and suspension systems together. The front uprights are perhaps one of the most complex parts on the car, and must be designed very carefully to make sure each system can mate and work together properly. First, the uprights house the wheel bearings, which then connect to the hubs and allow the wheels to spin. The bearings are held inside the bore of the upright by c clips which are recessed into the inner face of the upright bore. This secures the wheel bearing inside the upright and prevents it from slipping laterally, which would lead to the entire wheel falling off. Next, the uprights must also secure the brake calipers rigidly in the correct position to grip the brake rotors. The mounting tabs must be in the exact location that positions the brake rotor in the center of the caliper, and ensures that the caliper will not contact the rim or rotor as it spins. Lastly, the upright will connect to both arms of the suspension and the tie rods from the steering rack. For the rear uprights this connection is used to control the toe of the wheel to ensure it is near parallel with forward. These three joints use spherical bearings to allow extra degrees of freedom needed in the complex action of suspension travel. The relation of these mounting points is extremely important in how the car handles and turns.

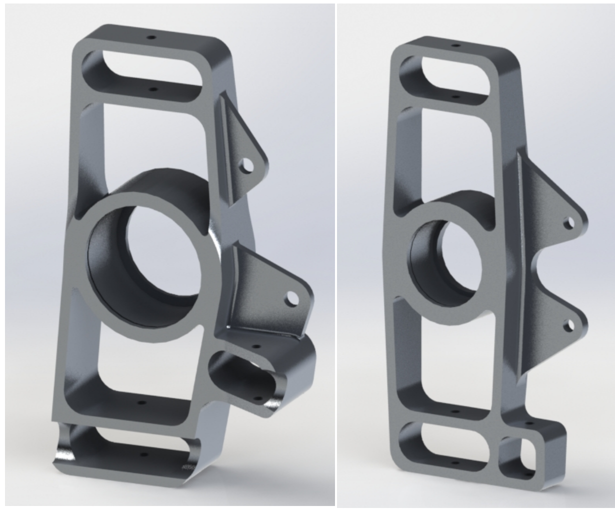


Figure 96 Front and Rear Upright

The three main geometrical relationships between the suspension mounting points are caster, kingpin inclination, and trail. Caster is the angle between the two ball joints from the vertical, when looking at the side view of the tire. The angle is always measured in the forward direction, and all cars will have some degree of positive caster. The effect of rotating the kingpin axis clockwise is that is causes the wheels to stay pointed forward when rolling and dampens the return of the wheel to center. This combined with trail is why a car will often track in a straight line instead of veering to one side without driver input. Trail is a measurement that is often caused by adding caster to an upright geometry, but it can also be increased or decreased manually. It is the measurement of the offset from the wheel center laterally to the kingpin axis when viewed from the side and is measured in the forward direction. Trail has the same effect as caster, but more directly related to returnability to center. Finally, KPI, or steering axis inclination is the angle between the ball joints from the vertical when looking from the front of the vehicle. This is measured from the upper ball joint to the wheel center, and is used to aid in returnability of the wheel and to reduce the scrub radius of the front wheels. The scrub radius is the distance between the tire center and kingpin axis when viewing from the front of the vehicle. Having a large scrub radius makes the car much more difficult to turn, because the wheels are being dragged around the kingpin axis. It is important to make the scrub radius as small as possible, in order to reduce fatigue on the driver when navigating through turns.

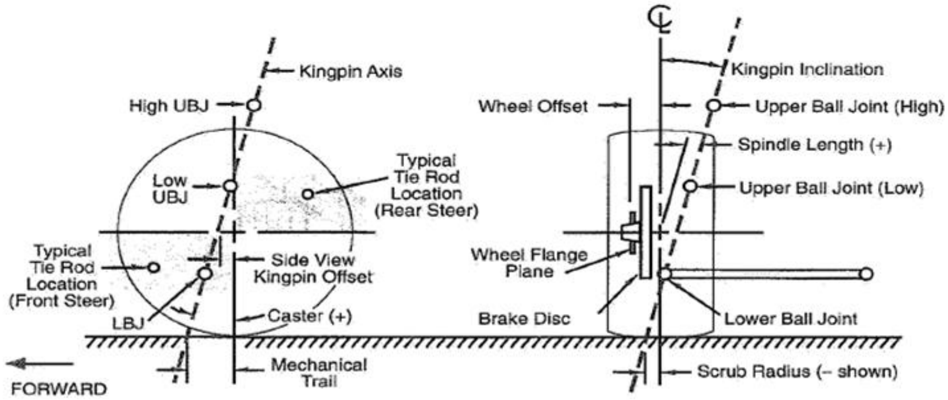


Figure 97 Upright Geometrical Components

After researching heavily on the FSAE forums and using production vehicles as an example, minor improvements were made on the upright configuration of the past vehicle. A caster angle of 5 degrees, a kingpin inclination of 2.3 degrees, and a mechanical trail of .5in were chosen. The resultant scrub radius was about 1.8 inches, which is about equal to that of the current car. This setup will reduce driver strain when navigating the tights corners of the endurance course and will keep the car tracking straight when not gripping the wheel.

Validation

To ensure that the uprights will survive the driving conditions seen in competition it was again necessary to run loading simulations on the parts. Again braking and bump situations were tested with extreme conditions. For the braking tests, the suspension and steering mounting points were all fixed along with the inner bearing housing to simulate the assembly of the car while a torque load was applied to the caliper mounting tabs. In the bump test, the bearing housing was fixed while upward forces were applied to the suspension mounting points. For the braking tests, both uprights had a factor of safety (FOS) of over 2.5, and for the bump tests the FOS was 12. This is a very good factor of safety for these components and is an improvement over the strength of the current car’s components.

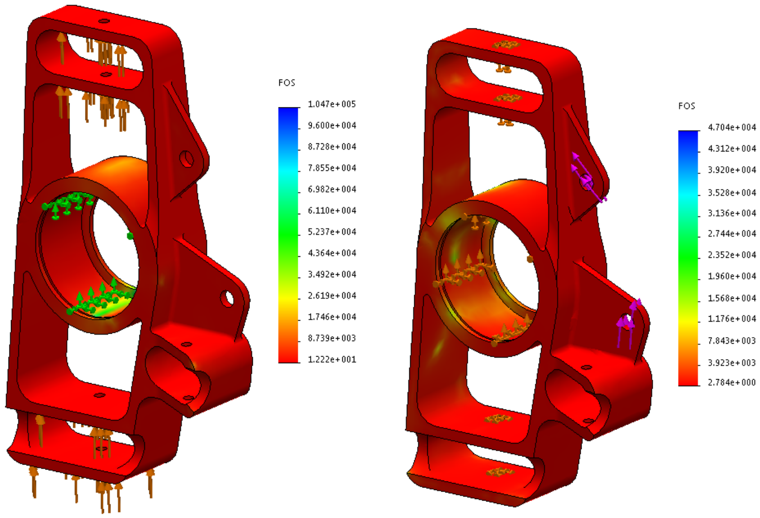


Figure 98 Factor of Safety Plots for the Front Upright

Electronics and Accessories

The accessories of the new FSAE car cover a wide variety. Some are accessories for the engine, while others are sensors and displays. Each has a specific purpose that is crucial to the function of the car.

Cooling System

The cooling system is extremely important to the function of a car. It maintains the temperatures of the engine within a safe operating range. If the temperature of the engine becomes too high, it can be dangerous because it can cause components of the engine to warp or break. Selecting the correct cooling system depends on the size and function of the engine. If a cooling system used on an engine is too small, it will not properly cool the engine and can lead to failure. On the other hand, it is less of a problem if a cooling system is too large for the engine. The downfall to having an oversized radiator is the space that will be wasted. Space has to be taken into account for many reasons. The pieces of an engine are intricately placed to minimize weight and the center of gravity. If the radiator is too large and is placed in such a way that it can disrupt other components of the car it becomes a problem. It is also important to consider the fact that a radiator needs to be placed in such a way that cooling air is always capable of flowing through its core. It should not be covered or blocked in any way. To enhance the cooling ability of the radiator, fans are attached to the radiator. Fans improve the circulation of the air through the radiator fins.

The radiator that was purchased for the 2016 FSAE car is from a YFZ350 Banshee. The new radiator is almost identical to the radiator that is on the current car. This is helpful to the SAE club members that will be working on the car because they can translate their familiarity of the older radiator into the assembly and attachment of the new radiator. According to the size of the engine, the radiator only needed to be 65 cubic inches. However, the radiator that was purchased is 155 cubic inches. This is not a problem for the car because it will not have a negative effect on the engine. Also, because of the turbocharger the additional cooling ability is welcome since an intercooler will not be installed. Ideally a turbocharger is accompanied by an intercooler but the turbocharger is small in comparison to other applications rendering a complete intercooler setup unnecessary. Although the radiator is larger than what is required for our engine size, it was substantially cheaper than other radiators.

Fuel System

The main components that make up the fuel system are the fuel tank, fuel pump, injectors, and regulator. According to the FSAE rules for competition, the engine must have electronic fuel injection (EFI) if boosted by a turbocharger. The engine was originally carbureted but it was converted to EFI, as stated previously.

After looking through previous FSAE competitions, it was determined that the fuel tank does not need to be larger than one gallon. This is good for the team because it means the team can use a smaller tank and not have as much trouble trying to fit it into the assembly of the car in such a way that any excess fuel left in the tank can be extracted. One part of the competition is to determine the fuel consumption of the car. The judges do this by measuring the beginning and ending volumes of fuel in the tank. In past WPI experiences, the judges have had trouble removing the fuel that remained in the tank, which affected the team's score. There are no restrictions on the type of fuel tank that must be used. The only FSAE rules regarding the fuel tank are that a filler neck must be present, there must be a way to see inside the fuel tank to see the fuel level, the tank must be inside the area of the car, and the tank cannot bear any weight. There are only a small amount of one gallon tank shape variations. This led to the idea of 3D printing our own tank. If this is done, the shape and location of the tank can be determined to specifically meet our needs. It also will not incur any purchasing or shipping costs specifically for the tank because it would be done on campus for no charge to the FSAE team.

One significant advantage of selecting a single cylinder engine is naturally high fuel efficiency. Reducing the weight of the car increases efficiency and the 19mm airflow restrictor affects single cylinders substantially less than 4-cylinders, forcing the latter to run at reduced power and efficiency. To determine the estimated sizing of the fuel tank, a study of past FSAE teams using a single-cylinder engine was performed to see the amount of fuel consumed. The ranking for fuel efficiency was also collected with surprising results: in 2012 only 1 team using a single cylinder engine placed outside the top 10. Therefore the car should score most of the 100 points available for the fuel efficiency scoring, despite using E85. In fact, the number 1 ranked team for fuel efficiency used a turbocharged single-cylinder engine with E85.

Team Number	Rank	Fuel Used (gal)
1	6	0.628
5	1	0.762
11	9	0.697
15	3	0.604
67	16	0.814
79	5	0.627

Table 24: Fuel Consumption and Ranking of Single Cylinder Teams in 2012

To determine the size of the fuel pump that was necessary for our engine, the team first looked at the size of the injectors to see how much fuel was required by our single cylinder engine. The fuel injector size was calculated with the following equation:

$$\frac{HP \times B.S.F.C.}{\#\ of\ injectors \times Duty\ Cycle} = \frac{65 \times 0.55}{1 \times 0.8} = 44.92\frac{Lbs}{Hr} = 472\frac{cc}{min}$$

Knowing that there is only one fuel injector because the engine only has one cylinder, it was known that this is the amount of fuel that must be supplied to the engine by the fuel pump. However, because this fuel system is being used for a racing application, the pressure that must be supplied is 43 psi. Also, because the engine will first run without the battery, the stator can only supply a small amount of current to run the fuel pump. This limitation is what added to the difficulty in finding a fuel pump that was able to fit our needs. The two options that were closest to being exact fits for our application were the Walbro GSL393 and the MSD 121-2225.

Exhaust System

The biggest restriction determined by FSAE rules is that the exhaust system cannot be louder than 110dBC at a 45 degree angle from the horizontal. This was our biggest concern when purchasing an exhaust system because the car must fulfill every rule to be eligible to compete. For sake of simplicity and ensuring that the muffler would be a proper fit to the engine, a stock YFZ450 muffler made of steel was purchased.

Intake System

According to the FSAE rules the intake system must have a limited intake diameter. The restrictor must be 19mm across when using E85 as the fuel. It is important to try to avoid turbulent flow as much as possible. This can most effectively be done by making the restrictor in the shape of a venturi nozzle. The pressure drop is smallest when keeping the difference in diameter relatively small, so that the difference in velocity is small and in turn the pressure drop is small as well. This can be seen when looking at the Bernoulli equation and flow rate. Flow rate is calculated with the diameter of the pipe and the velocity of the fluid.

$$Q = A_1V_1 = A_2V_2$$

$$V_2 = V_1\frac{A_1}{A_2}$$

The Bernoulli equation shows the relationship of the difference in pressure with respect to the difference in velocity.

$$P_1 + \frac{1}{2}V_1^2 = P_2 + \frac{1}{2}V_2^2$$

$$\Delta P = \frac{1}{2}[V_2^2 - V_1^2]$$

$$\Delta P = \frac{1}{2}\left[\left(V_1\frac{A_1}{A_2}\right)^2 - V_1^2\right]$$

From this, the ratio of the diameters and the velocity of the air entering the system will determine the difference in pressure of the system.

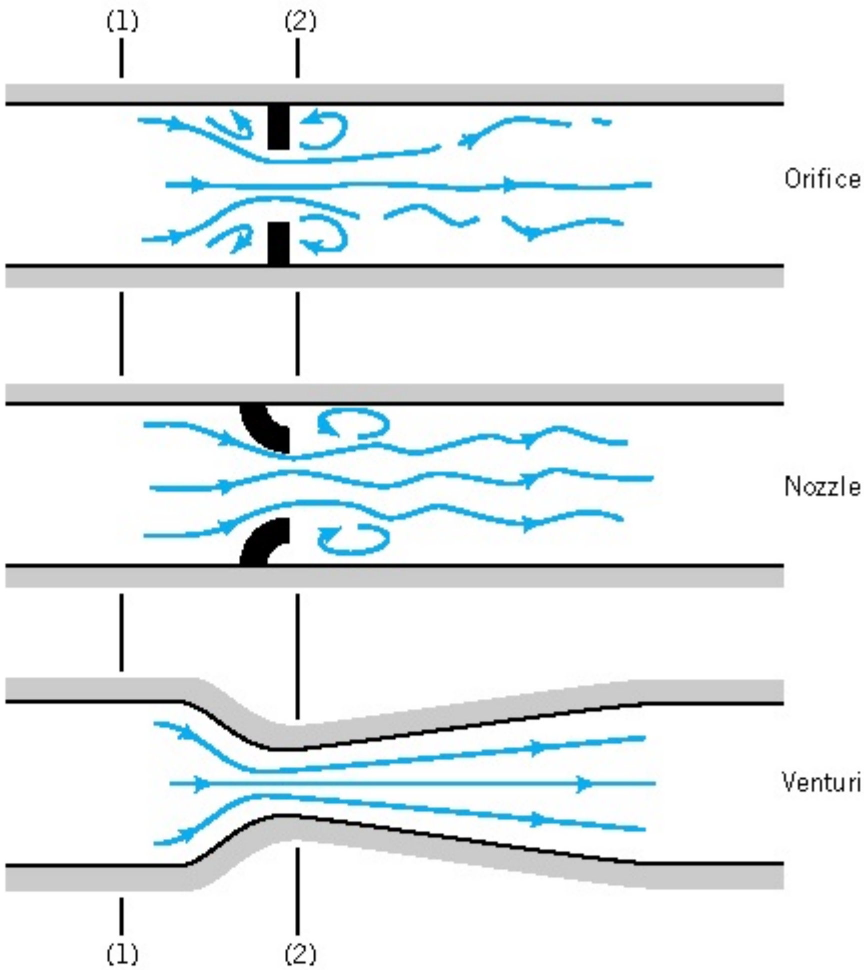


Figure 99: Pipe Flow Comparison

Electronics

The engine control unit (ECU) that is used on the current car is a Haltech Platinum Sport 2000. There is another unused Haltech Platinum Sport 2000 available for use, but it is preferred to sell it and purchase a Haltech Elite 1500 instead. The Elite offers better performance and more advanced programming. It is also protected in a durable waterproof case that will prevent damage reliably. It comes with electrical relays and fuses that would otherwise need to be purchased, reducing the price differential between the versions to a reasonable figure.

There are multiple sensors that the FSAE team would like to display. Some of these are oil pressure and temperature, engine temperature, rpm, among others. To display these the team will be utilizing an Android wireless device. Through Bluetooth connectivity and an OBD2 transmitter, an app on the android device can serve as a dashboard displaying all of the information the team would like to see. This is important because the driver will be able to have real time data when driving the car. Since the car is being built to attend the competition, competitiveness must be achieved in all aspects. When tuning the car, it is important to understand what is functioning well and what needs work. This display system can offer the team very important information.

The electronic portion of the car is very complicated because the group needs someone with experience in electrical and computer engineering. There are many things that also need to be purchased because although the Haltech will be recycled, the display along with many sensors and wiring kits need to be purchased.

Future Recommendations

To further assist next year's team in successfully completing the competitive FSAE car there are recommendations that will help expedite the completion of the car.

In preparation for a long year of hard work there are things that should have an urgency to them. First, try to get the engine running as quickly as possible. The engine is the heart of the entire car. If the engine is no good, the rest of the car will be irrelevant. It is better to understand how the engine performs early on in the year because if there are any problems that arise, there will be enough time to correct them with well thought processes instead of simply doing things that may fix one problem but due to the lack of quality may cause other problems. Secondly, manufacture the uprights as early on in the year as possible. These are crucial to the performance of the car and have to be exact to perform the way it was intended. By manufacturing the uprights early, there will be enough time in the year to make changes, if it were necessary to do so.

Also, there are certain components of the car that may be forgotten altogether. These are not the major subsystems of the car but instead these may be things that are specific to the FSAE rules. It is recommended that more than one person check the rules explicitly to ensure that the final product will be allowed to compete. If at the end of the year the final product is not allowed to compete because of a small error, hours of hard work will have been done in vain. Therefore, as it was done with the frame this year, have more than one person double check every rule.

Partnership with SAE Club

The relationship between the FSAE MQP and SAE club has always been somewhat confused and underutilized. Typically the MQP does all of the design work, works with SAE to manufacture and assemble the car, then leaves the car for SAE to take to competition. One of the primary goals of this MQP was to change this relationship and integrate more fully with SAE club by bringing them into the design process as well. This creates a virtuous cycle of being able to design more with the added research and people, SAE club being more familiar with the car and maintain and improving it, and recruiting knowledgeable candidates for the next MQP. This was acheived by going to SAE meetings and making our intentions to work with any interested members well-known and dividing into and leading our subgroups: chassis and suspension, drivetrain, brakes and steering, and accessories and electronics. Each subgroup held weekly meetings and twice per term all groups met to review the entire process and make decisions on the designs selected by the subgroups. Many group members were new to the technical information presented and gave us an opportunity to teach. Although attendance started very strongly and dropped as the terms progressed and other time commitments came to light, there was a group of about 10 additional members who are working very closely with the team. This relationship will only grow stronger with time, especially as the amount of hands-on work increases and students can see the fruits of their labor in a functioning FSAE car.

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Appendix

Appendix A

Assumptions							
Total Weight	450	lbs	Weight Distribution:				
Front Weight	225	lbs	Front	50%			
Rear Weight	225	lbs	Rear	50%			
CG Height	12	in					
Wheelbase	61	in					
Max Braking Force	1.5	g	also used as tire firction coefficient				
Weight Transfer							
M*a*CG Height/WB Length							
WT	132.7868852	lbs					
Dynamic Front Weight	357.7868852	lbs	80%				
per wheel	178.8934426						
Dynamic Rear Weight	92.21311475	lbs	20%				
per wheel	46.10655738						
REAR					**FRONT**		
Friction lock up force=u*Normal Force					Friction lock up force=u*Normal Force		
u_{tires}	1.5				u_{tires}	1.5	
lock up rear per wheel	69.15983607	lbf			lock up front per wheel	268.3401639	lbf
Torque=force*distance					Torque=force*distance		
rear wheel torque	59.07402664	ft lbs			front wheel torque	229.2072234	ft lbs
also equal to rotor torque					also equal to rotor torque		
Rotor and pad Friction force					Rotor and pad Friction force		
Friction force=torque/rotor radius	177.2220799	lbf			Friction force=torque/rotor radius	687.6216701	lbf
Clamping Force					Clamping Force		
Clamping Force=friction force/u of brak	443.0551998	lbf			Clamping Force=friction force/u of	1719.054175	lbf
Caliper force					Caliper force		
clamping force/2	221.5275999	lbf			clamping force/2	859.5270876	lbf
Line Pressure (also master cylinder pressure)					Line Pressure (also master cylinder pressure)		
Caliper force/caliper area	280.4146834	psi			Caliper force/caliper area	358.1362865	psi
Brake pedal output force					Brake pedal output force		
line pressure*master cylinder area	168.5292247	lbf			line pressure*master cylinder area	215.2399082	lbf
Brake pedal input force					Brake pedal input force		
output force*pedal ratio	28.08820412	lbf			output force*pedal ratio	35.87331803	lbf
Total input force to lock wheels	**63.96152215**	**lbf**					
Balance (Rear, Front)	44%	56%	can be adjusted to 50% by balance bar				

Caliper	Rotor Sizes	Piston Area	Piston Size (dia)	Weight	Pad Area	Cost
GP200	11 max	1.23	1.25	0.9	unlisted	$ 98.93
Dynalite Single IIIA	13 max	2.4	1.75	1.8	3	$ 88.50
PS-1	6-9in	0.79	1	0.93	2	$ 91.74
Billet Dynalite Single	13 max	2.4	1.75	1.58	3	$ 114.57
GP320	9-11.5	2.46	1.25 x2	1.7	3.65	$ 178.19

Master Cylinder Sizes

Diameter (inches)	Area (inches^2)
0.625	0.306796158
0.75	0.441786467
0.8125	0.518485506
0.875	0.601320469
1	0.785398163
1.125	0.994019551

Component	Quantity	Supplier	Model	Weight	Cost	Material	Size	important specs
Caliper, Rear	2	Wllwood	PS-1	0.93	$ 183.48	Cast Aluminum		
Caliper, Front	2	Wllwood	Dynalite Single	1.8	$ 229.14	Billet Aluminum		
Rotor and tabs	4	RCV	220mm rotor kit		$ 432.00	Steel	8in	
Master Cylinder Front	1	Wllwood	Compact Remote Flange Mt Master Cyl-1/8 NPT Outlet	Unlisted	$ 85.95			
Master Cylinder Rear	1	Wllwood	Compact Remote Flange Mt Master Cyl-1/8 NPT Outlet	Unlisted	$ 85.95			
Pedal Assembly	1	Wllwood	Floor Mounted Brake Pedal		$ 125.63		6:1 ratio	
Brake Lines	2							
Flex Line	4							
Fittings								
Tie Rods	2							
Steering joints	2							
Steering Rack	1							
Steering Shaft	1							
Steering U Joint	1							
Upper steering shaft	1							
Quick Release Wheel Fitting	1							
Uprights, Front	2	Custom						
Uprights, Rear	2	Custom						
Hubs, Front	2	Custom						
Hubs, Rear	2	Custom						
Wheel Bearings, Front	2				$ 100.00			
Wheel Bearings, Rear	2	Partzilla	30x54x24 bearing		$ 64.30		30x54x24	http://www.partzilla.com/parts/detail/honda/HP-91051-HA7-651.html
C Clips, Rear	2	Partzilla	inner circlip 55mm		$ 3.92			http://www.partzilla.com/parts/detail/honda/HP-94520-55000.html
C Clips, Front	2							
Castle Nuts, Front	2							
*****Add Shocks					$1,310.37			

Braking calculations.xlsx

Appendix B

Equation for Calculating Car Radiator Size

Start with 2 cubic inches of core for every cubic inch of engine. Increase or decrease that value by the following factors:

ADD

- **0.1 for a vertical flow radiator core**
- **0.1 for an in-line engine**
- **0.1 for a small trailer towing**
- **0.1 for a 2 row radiator**
- **0.1 for double evaporators**
- **0.2 for outside temperatures of 105°F (40.5°C)**
- **0.2 for a medium trailer towing**
- **0.2 for a small engine fitted to a heavy car**
- **0.2 for a radiator fan with diameter less than 90% of smallest dimension**
- **0.3 for air conditioning**
- **0.3 for no fan shroud**
- **0.3 for an antique car with small engine compartment**
- **0.4 for large trailer towing**
- **0.6 for a diesel engine**

SUBTRACT

- **0.1 for remote transmission cooler (not within radiator)**
- **0.1 for standard in-line transmission**
- **0.1 for a single row radiator**
- **0.1 for a V6 / V8 engine**
- **0.2 for a spacious pickup truck engine compartment**
- **0.2 for outside temperatures less than 90°F (32.2°C)**
- **0.2 for a full fan shroud**
- **0.2 for a horizontal flow radiator core**
- **0.3 for a large engine in a small car**

$$2 + 0.1 + 0.3 - 0.1 - 0.2 = 2.1$$

$$500cc = 30.512ci$$

$$2.2 \times 30.512 = 67.1264ci$$

Appendix C

Vsusp web suspension HTML save file.

http://vsusp.com/#0.8%26project_name%3A2015%20WPI%20FSAE%26trim%7Bbody_roll_angle%3A0
%7Cfront.left_bump%3A0%7Crear.left_bump%3A0%7Cfront.right_bump%3A0%7Crear.right_bump%3A
0%7D%26front%7Bframe.susp_type%3A0%7Cframe.bottom_y%3A13969%7Cframe.center_to_upper_m
ount_x%3A29209%7Cframe.bottom_to_upper_mount_y%3A17780%7Cframe.center_to_lower_mount_x
%3A24130%7Cframe.bottom_to_lower_mount_y%3A0%7Ccontrol_arms.upper_length%3A26923%7Cco
ntrol_arms.lower_length%3A32151%7Cknuckles.hub_to_upper_x%3A8450%7Cknuckles.hub_to_lower_
x%3A8450%7Cknuckles.hub_to_lower_y%3A11430%7Cknuckles.hub_to_upper_y%3A11430%7Cknuckl
es.hub_to_strut_axis%3A14000%7Cknuckles.strut_incl%3A8000%7Cwheels.offset%3A1270%7Cwheels.
diameter%3A1500%7Cwheels.diameter_expl%3A35560%7Ctires.size_convention%3A1%7Ctires.section
_width%3A19500%7Ctires.aspect_ratio%3A4500%7Ctires.diameter_expl%3A52069%7Ctires.width_expl
%3A17780%7Ctires.compression%3A0%7D%26rear%7Bframe.susp_type%3A0%7Cframe.bottom_y%3
A13969%7Cframe.center_to_upper_mount_x%3A25400%7Cframe.bottom_to_upper_mount_y%3A1778
0%7Cframe.center_to_lower_mount_x%3A20320%7Cframe.bottom_to_lower_mount_y%3A0%7Ccontrol
_arms.upper_length%3A30703%7Ccontrol_arms.lower_length%3A35760%7Cknuckles.hub_to_upper_x
%3A8823%7Cknuckles.hub_to_lower_x%3A8823%7Cknuckles.hub_to_lower_y%3A10160%7Cknuckles.
hub_to_upper_y%3A10160%7Cknuckles.hub_to_strut_axis%3A14000%7Cknuckles.strut_incl%3A8000
%7Cwheels.offset%3A1270%7Cwheels.diameter%3A1500%7Cwheels.diameter_expl%3A35560%7Ctire
s.size_convention%3A1%7Ctires.section_width%3A19500%7Ctires.aspect_ratio%3A4500%7Ctires.diam
eter_expl%3A52069%7Ctires.width_expl%3A17780%7Ctires.compression%3A0%7D%26pref%7Bdiag1.
px_per_mm%3A200%7Cdiag1.front_or_rear%3Arear%7Ctab.active%3A3%7Cunits%3A0%7Cshow.f%3
A1%7Cshow.ca%3A1%7Cshow.k%3A1%7Cshow.w%3A1%7Cshow.t%3A1%7Cshow.rc%3A1%7Cshow.
ic%3A1%7Cshow.fvsa%3A1%7Cshow.tl%3A1%7Cshow.kpil%3A1%7Credraw_during_drag%3A1%7Cch
art.x_axis_center%3A0%7Cchart.x_axis_window%3A10%7Cchart.x_axis_num_steps%3A10%7Cchart.x
_axis_field%3Atrim.%5BFR%5D.right_bump%7Cchart.y_axis_fields%3A%5BFR%5D.general.roll_center.
x%7D